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A CONCISE TREATISE ON THE THEORY AND
PRACTICE OF LUBRICATION;
THE PHYSICAL, CHEMICAL, AND MECHANICAL
PROPERTIES AND TESTING OF LIQUID AND
SOLID LUBRICANTS

WITH NOTES ON RECENT DEVELOPMENTS AND
EXAMPLES FROM PRACTICE

FOR ENGINEERS, CHEMISTS, AND STUDENTS

BY

J. H. HYDE
A.M.INST.C.E., A.M.I.MECH.E., M.I.A.E.

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PREFACE

WHILST the subject of Lubrication and Lubricants cannot be dealt with exhaustively in a small book, there is undoubtedly a demand for a compact work for the use of engineers and chemists, and of engineering students in particular, which will indicate the present state of knowledge on the subject, the experimental and other work by which it has been obtained, and the efforts which are being made to increase it.

At the present time, very considerable attention is being given to the study of lubrication generally, not only by oil chemists and engineers, but also by workers in the field of "pure science."

In the following pages, endeavour has been made to present the subject to the student in as simple a manner as possible, consistent with the assumption of a general knowledge of engineering terms in common use, and to indicate to him, broadly, the course which his studies should pursue.

The best thanks of the author are due to Mr. T. Rooney for his suggestions in connection with the chapter on "Chemical Tests."

J. H. HYDE.

TEDDINGTON,
MIDDLESEX.

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LUBRICATION AND LUBRICANTS

CHAPTER I

FUNCTION, TYPES, AND APPLICATION OF LUBRICANTS

Function of Lubricants. When two bodies are pressed together, the resistance offered to the sliding of one over the other is termed " frictional " resistance. In machinery and mechanism of all kinds, this frictional resistance to motion has the ultimate effect of reducing considerably the useful work done, thereby decreasing the utility and efficiency. The term " lubrication " is applied to the various methods of reducing the amount of frictional loss by the interposition between the rubbing surfaces of a film of greasy substance termed the " lubricant."

The efficiency of the lubrication depends upon many factors connected with the lubricants and also, in a considerable degree, upon the method by which it is applied.

When the pressure between the rubbing surfaces is great, and the relative velocity high, the presence of lubricant is absolutely essential, for otherwise the bearings of the machine in which such conditions obtain would rapidly be destroyed either by the heat generated or by " seizure " of the bearing surfaces.

It will be shown later that the lubrication of metal surfaces is usually effected by substituting for the "solid" surface friction the "internal" friction of a fluid or plastic substance which, under the same conditions of load, speed, and temperature, is subject to different laws, and is usually of a much smaller magnitude than solid friction.

Types and Choice of Lubricants. The lubricants in common use are either in the form of a fluid or in a plastic condition, and are of two main classes, viz., "fixed" oils and fats and "mineral" oils and greases. For certain purposes solid lubricants are used, but these are mostly less efficient than fluid substances.

The fixed oils are derived from vegetable and animal sources, and are usually excellent lubricants, but suffer from certain disadvantages in that they possess properties of "gumming," which results in drying of the oil when in a thin film and in etching, or corrosion of metallic surfaces due to the presence of free acid.

The comparatively recent introduction of mineral lubricating oils which do not gum or corrode has led to their very general use in spite of their inferiority as lubricants in comparison with the fixed oils.

The choice of a lubricant most frequently depends upon the ease with which it is obtained and on the cost. At the present time, enormous quantities of mineral oils of all kinds are produced and can be obtained generally at a lower cost than the fixed oils; hence their almost universal use for ordinary purposes.

Lubrication. Generally, it would appear that future improvements in lubrication, i.e. the more efficient application of lubricants by avoiding waste and effectively reducing frictional losses, are of equal if not of greater importance than the improvement of lubricants by so mixing them or purifying them in order to obtain a slightly better lubricating value.

Immense quantities of lubricants are wasted due to inefficient application and, although engineers have made great strides in lubrication since the days of the important experiments of Beauchamp Tower (1885), yet there is undoubtedly room for very much improvement in methods generally. Advance in lubrication has not kept pace with the development of machinery; in fact, limits in sizes and weights are often determined by the system of lubrication.

Since materials of high tenacity have come into more general use, the reduction in size of parts, which is thereby rendered possible, is somewhat restricted in cases where large bearing surfaces have to be retained in order to provide for adequate lubrication.

To the student of lubrication, a considerable knowledge of engineering is essential in order that he may understand and appreciate the difficulties which exist in efficiently lubricating machines under varying conditions, and the manner in which those difficulties may be overcome by a modification of design or of the system of application of the lubricant.

CHAPTER II

FRICTION

Static and Kinetic Friction. If a slab *A* (Fig. 1), of weight *W*, is laid on a level plate *B*, the force *F* applied to *A*, which is just sufficient to cause it to slide, is a force which is equal to, but which acts in the opposite direction from, the frictional resistance. The force of friction itself acts in the plane of the surfaces as indicated at F_1 (Fig. 1), and experiment will show that it is proportional to the weight *W* or, more exactly, the normal force between the surfaces. The amount of frictional resistances per unit of normal force, or the ratio of frictional force to normal pressure, is termed the "coefficient of friction," and is thus represented by

$$\frac{F_1}{W} = \frac{F}{W} = \mu$$

The force required to start the motion of the slab usually differs from that required to maintain the motion, and is most frequently of a greater amount. The coefficient of friction at starting is termed the "static" coefficient, and that when the motion is maintained the "kinetic" friction.

The kinetic or running friction is the resistance with which lubrication has chiefly to deal, but in certain cases the static friction cannot be neglected, although generally the vibration or slight slackness or elasticity of parts of a machine at starting will considerably reduce its effect.

Variation of Friction with Nature of Surfaces. The amount of frictional resistance varies extensively between surfaces which are soft and rough and those which are hard and polished, being greater between the former than the latter.

When soft and rough, the surfaces " bed " into each other and, if the normal pressure is great, the sliding causes abrasion, which does not necessarily result in a smoothing of the surfaces.

The friction being least when the surfaces are hard and polished, such surfaces are usually adopted in practice. On the other hand, very smooth, flat, and clean surfaces, when pressed together, have a tendency to adhere to each other, sometimes in such close association that the frictional force is enormously increased. If relative motion is produced under these circumstances, the surfaces seize together and, if the motion is continued, abrasion takes place to such a degree as to render them useless for bearing surfaces.

The frictional force between dissimilar metallic surfaces is invariably less than that between similar surfaces; and, although the results of various experiments on this point do not agree, the disagreement probably arises from the fact that the surfaces experimented upon were not always chemically clean, a fact which later research has demonstrated to be of utmost importance. Mere wiping of the surfaces is not sufficient to render them perfectly clean, as any surface is most easily contaminated by a film of grease or by a very thin layer of oxide or other product of the chemical action of moisture and active vapours suspended in the atmosphere. The effect of the contaminating film is to modify considerably the frictional resistance between the surfaces, and unless the greatest care be taken to remove it, the true value of the coefficient of friction cannot be obtained.

With clean metallic surfaces, seizing takes place very readily even under small pressures; but cast iron, possibly because of minute particles of carbon acting as a lubricant, does not easily seize at moderate loads and speeds.

Heat Developed by Friction. The work expended in producing relative motion between two surfaces,

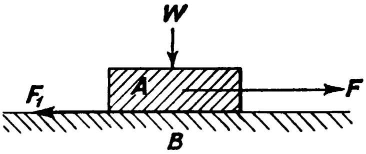

Fig. 1.—Illustrating Frictional Resistance to Sliding.

such as those indicated in Fig. 1, is measured by the product of the frictional force and the distance moved through by one surface relative to the other. The work appears as heat, which has the effect of raising the temperature of the parts. The temperature rise depends upon the rate at which the work is expended, and on the rate at which heat is conducted through and radiated from the mass of metal adjacent to the surfaces. If the friction is very great, the temperature may rise to such an extent as to overheat or even melt the surfaces in contact.

Variation of Friction with Load and Speed. Experiments have shown that the coefficient of friction between clean surfaces varies approximately in proportion to the total normal pressure, does not vary with speed for low pressure and speed, but increases when the speed and pressure are high. It also depends in a considerable degree upon the nature and condition of the opposing surfaces, and does not appreciably vary with temperature, provided the heat is insufficient to change the condition of the surfaces in contact.

It should be clearly understood that the frictional resistance is a function of the normal load and not of the intensity of the load per unit area of surface in contact ; in other words, the friction is independent of the area of the surfaces in contact.

Rolling Friction. When a cylinder rolls over a flat surface, the contact is theoretically along a generating line of the cylinder. Practically, however, the weight of the cylinder, together with the load it may support, causes a deformation of both surfaces. This deformation occurs in a greater or lesser degree with any elastic material, and the cylinder or roller sinks into the plate until the area of contact is sufficient to support the load. Obviously, if the contact were along a line, the intensity of pressure would be infinite.

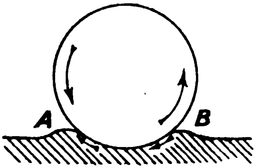

FIG. 2.—DEFORMATION RESULTS IN ROLLING FRICTION.

Referring to the sketch in Fig. 2, which illustrates the deformation of a soft plate by a hard roller, the material *A* forced up in front of the roller slides over the latter in a direction contrary to the direction of motion of its periphery. At the same time, the mass *B*, due to the elasticity of the material, in returning to its original form, slides forward with respect to the roller, and again in a direction contrary to the periphery. Thus the friction brought into play in both cases impedes the motion of the roller, and the laws of friction for dry surfaces are applicable.

Ball Bearings. In ball bearings, endeavour is made to keep the curvature of the ball races as nearly equal to that of the balls as practicable. When the load is applied, the balls tend to flatten at their points of contact with the races, and the distortion which takes place depends on the relative curvature of the ball and race. If the deformation is excessive, the balls fail by their outer surfaces " shelling " off and, once this begins, the bearing is rapidly destroyed.

The lubrication of ball bearings is discussed in Chapter VIII.

Fluid Friction. The internal friction or viscosity of a fluid is the resistance set up to the free motion of the particles of the fluid over each other. It may otherwise be described as the resistance which the fluid offers to shearing forces. Maxwell's definition is ; " The viscosity of a fluid is measured by the tangential force on unit area of either of two horizontal planes at unit distance apart, one of which is fixed, while the other moves with unit velocity, the space between the surfaces being filled with the viscous substance."

The property of viscosity is possessed by all fluids in varying degree, as may be observed by the ease or otherwise with which they flow : thus, a certain volume of rape oil, at atmospheric temperature, will take over 100 times as long to flow from an orifice as an equal volume of water at the same temperature.

Consider two flat plates between which a fluid has been introduced, the lower plate being fixed and the other moving with a velocity v in a direction parallel to its surface. There will be no flow of fluid relative to either plate in the immediate vicinity of its surface, i.e. the fluid at the surface of the fixed plate will be at rest, while that at the surface of the moving plate will have a velocity equal to v.

In the space between the two plates, the fluid may be considered as being made up of superposed layers, and each layer will be moving at a velocity proportional to its distance from the fixed plate. As the fluid possesses viscosity, a continuously applied force is necessary to maintain the constant velocity v of the upper plate, and this force is transmitted to the lower plate through successive layers of fluid.

Let the total force producing motion be P, the area of the plate a, and the distance between the plates x.

Then f, the intensity of the shear stress, is given by $f = P/a$; and, because in one second the distance travelled by the upper plate relative to the lower is equal to v, the shear strain is equal to v/x and is constant throughout the distance x.

Now v/x is termed the rate of distortion, and the stress required to produce unit rate of distortion is called the coefficient of viscosity and is usually indicated by η. Thus—

$$f = \eta \frac{v}{x} \text{ or } \eta = \frac{f}{(v/x)}$$

and for unit velocity v and unit distance x, η becomes equal to f.

This is the true conception of viscosity, and it will be noted that it is a stress and is measured in stress units, i.e. in dynes per square centimetre.

Viscous flow between planes is very important in lubrication, but too many practical difficulties are involved to enable it to be used as a method of determining the value of η for a fluid, and consequently the determination is usually made by estimating the rate of flow of the fluid through a capillary tube of known dimensions. It is not within the scope of this book to enter into a discussion on the theory underlying the flow of a fluid through capillary tubes; it will be sufficient for the purpose in hand to give the results obtained, assuming lamina flow and absence of slip at the boundaries as in the case of parallel planes.

It has been shown that the coefficient of viscosity varies directly as: (i) the fourth power of the radius, r, of the capillary tube; (ii) the head, h, of fluid producing the flow; (iii) the density, ρ, of the fluid; and the time, t/V, taken for the efflux of unit volume; and, inversely, as the length, l, of the capillary tube.

If g represents the acceleration due to gravity, the coefficient of viscosity is expressed as—

$$\eta = \frac{\pi \rho g h r^4 t}{8 v l}$$

Osborne Reynolds has demonstrated that there exists a certain critical velocity above which the flow changes in character from lamina to sinuous or eddying motion in which the resistance to flow is proportional to the square of the velocity. He also found that for lamina flow the following equation must be satisfied—

$$\frac{g\rho^2 hr^3}{8l\eta^2} > 700$$

In constructing a viscometer, the figure 700 is usually considerably exceeded.

The viscosity of a fluid decreases with rise of temperature, and in the case of oils the decrease is frequently extremely rapid. Fig. 3 (p. 12) shows the variation with temperature for various oils of animal, vegetable, and mineral origin. The necessity for maintaining uniform and steady temperature throughout the oil, when making viscosity measurements, will be apparent on examination of these curves.

Experiments* carried out by the author have shown that the viscosity of lubricating oils is much affected by pressure. Increase in pressure is accompanied by a rise in viscosity corresponding, at 6 tons per sq. in., to four times the viscosity at atmospheric pressure in the case of vegetable and animal oils, and to anything between 8 and 20 times the low pressure value of viscosity for oils of mineral origin.

This increase is of importance, as in practical lubrication oil films are sometimes subjected to pressures of this order.

---

* *Proc. Royal Socy.*, Vol. xcvii, 1920.

Internal Friction of Semi-solid Substances. A semi-solid substance may be defined as a substance which necessitates a definite stress to produce in it a continuous change of shape. A true fluid, although perhaps exceedingly viscous, will begin and continue to flow under the smallest stress. To produce a change of shape in a semi-solid substance, however, requires a definite stress. As examples, tallow, being a soft solid, will resist a small shear stress for an indefinite time; pitch, however, which is almost a true fluid, will not do so, as may easily be demonstrated by placing a small weight on the surface

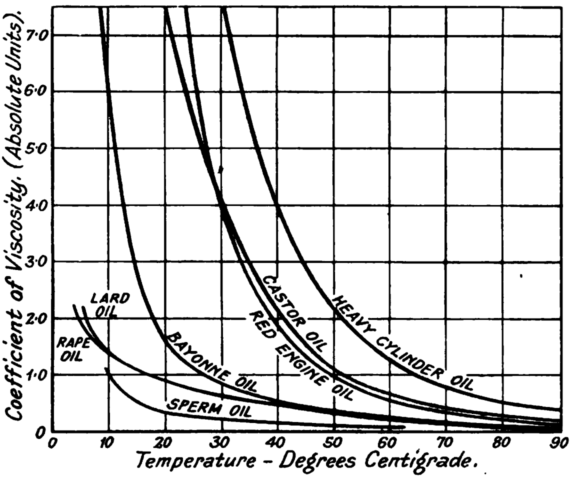

FIG. 3.—VARIATION OF VISCOSITY WITH TEMPERATURE.

of hard pitch contained in a barrel, when it will be observed that in time the weight will begin to sink and will finally disappear beneath the surface.

In lubricating with semi-solids, it would appear that the resistance to motion at starting would be considerable owing to solid friction, but that once the motion is begun the friction will approximate to that of a very viscous fluid.

CHAPTER III

THEORY OF LUBRICATION

WHEN two lubricated-bearing surfaces are pressed together, the film of liquid between them is gradually squeezed out and the surfaces approach one another. If, however, the pressure exerted is not too great, the surfaces will be prevented from coming into contact by the properties which the liquid film possesses of intimately associating itself with the surfaces and entirely wetting them, and by the resistance of the film to rupture.

When the load is increased sufficiently, rupture of the film takes place, the surfaces come into close contact, and any attempt to slide one over the other then results in seizing.

On the other hand, if the load is such as to allow the retention of a perfect film of lubricant, the resistance to relative motion depends on the viscosity of the liquid film.

Lubrication of Plane Surfaces. *Case I. Parallel Surfaces.* In Fig. 4, the surfaces AB and CD are assumed to be of infinite area and to remain parallel to each other, the upper surface, AB, being fixed and the lower surface, CD, moving with a velocity corresponding to $b'c'$. If there is a lubricating film between the surfaces, the boundary of the film in contact with the lower one will be moving at the same velocity as that surface, and the boundary in contact with the opposing surface will be stationary. The velocity distribution across the film may, therefore, be represented by the line *Ac′*; and the velocity of any point in the film, such as *a*, be represented by *aa′*.

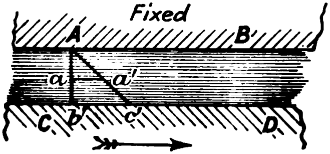

Fig. 4.—Lubrication of Parallel Plane Surfaces of Infinite Extent.

In such a case, the resistance to relative motion is directly proportional to the viscosity of the lubricant and to the velocity, and inversely proportional to the distance between the surfaces.

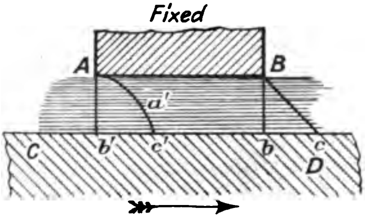

Fig. 5.—Lubrication of Parallel Plane Surface of Finite Extent.

If now, the upper surface is of finite dimensions (Fig. 5), the fluid will approach the edge, *A*, of the fixed surface at a velocity equal to that of the moving surface and, because of the inertia of the fluid, the distribution of velocity across the film between the surfaces at *A* is represented by the curve *Aa′c′*. At the edge *B*, however, the viscosity of the fluid will have asserted itself and the velocity distribution may again be represented by the straight line *Bc*.

The volume of fluid flowing into the space between the surfaces is proportional to the area of the figure *Aa′c′b′ A*, and that flowing out is proportional to *Bcb*; the difference between these areas represents the excess fluid which escapes in a lateral direction from both sides.

The inertia effect of the fluid entering at the edge *A* produces a force tending to separate the surfaces and tends also to the maintenance of the oil film.

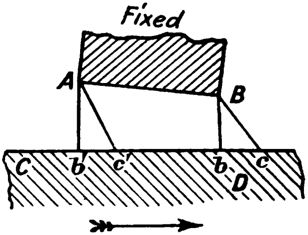

Fig. 6.—Lubrication of Inclined Plane Surfaces.

Case II. Inclined Surfaces. In Fig. 6, *C* represents the moving surface and *AB* the fixed surface as before; but, in this case, *AB* is inclined to the lower surface.

Reasoning as before, it will be seen that, even neglecting inertia, the volume of oil entering at *A* is greater than that leaving at *B*, and the excess fluid escapes laterally. The excess fluid, being viscous, resists expulsion at the sides, and a pressure is produced which forces the surfaces apart.

Osborne Reynolds has investigated mathematically the pressure distribution of the fluid on the upper surface, and has shown that the centre of pressure lies nearer the edge *B* than the edge *A*.

Journal-bearing Lubrication. Beauchamp Tower's experiments * demonstrated for the first time the necessity of applying the lubricant on the unloaded side of the journal. This method allowed a perfect oil film to form between the bearing and the journal, and the frictional resistance was found to be considerably less than was the case when the oil was introduced through a hole near the point of maximum loading of the bearing, which was the common practice at that time.

Tower also found that when a bearing had been running for some considerable time, the bearing brass became worn to a greater radius of curvature than the journal, and that when such wear had taken place, the frictional resistance was reduced and a greater load could be carried.

Osborne Reynolds attributed the greater capacity for loading to the wedging action of the lubricant described in connection with inclined plane surfaces.

Tower's experiments were made on a journal 4 in. diameter, and the load was taken by a half bearing; it was arranged that the lower side of the journal ran in a bath of oil. Tests were made at loads up to 500 lbs. per sq. in. and at speeds ranging from 100 to 450 r.p.m. The results of the tests showed the frictional resistance to be approximately proportional to the speed, to vary considerably with the temperature of the bearing, but to remain constant for all loads up to the seizing load.

The existence of a pressure film of lubricant entirely separating the shaft and bearing was established, and Beauchamp Tower measured the intensity of oil pressure at various points and found that the total vertical pressure was equal to the load applied.

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* *Proc. Inst. Mech. Engrs.*, 1883; and *Phil. Trans.*, 1886.

Osborne Reynolds applied hydrodynamical equations to this case of lubrication, and evolved an equation giving the relation between the relative velocity of the bearing surfaces, the distribution of the oil pressure, and the thickness of the oil film. The results calculated from his equations agree very well with those found by experiment.

Reynolds also obtained the result that the film was of the same average thickness for all moderate loads and, when the pressure of the oil film was everywhere positive, the film thickness was equal to half the difference in radii of journal and bearing; and, further, that when the load was sufficient to decrease the film thickness below this value, the oil pressure on the "off" side of the bearing became negative and the film tended to break up.*

W. J. Harrison † has investigated mathematically the case of a journal bearing in which the brass completely encircles the shaft, and he has shown that when the minimum thickness of the oil film is equal to half the difference in radii of the journal and bearing, the moments on the shaft and bearing are unequal, being in the ratio of 2 to 1. It follows from this that the accuracy of the values of coefficient of friction between surfaces obtained in testing machines in which a complete bearing is used, is open to question.

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* The part of the bearing which a point on the shaft traverses in passing from the unloaded to the loaded side is termed the "on" side of the bearing: the other half of the bearing is termed the "off" side (*see* Fig. 8, p. 25). The division between the two sides is defined by a plane passing through and parallel to the axis of the shaft in the direction of the load.

† "The hydrodynamical theory of the lubrication of a cylindrical bearing under variable load, and of a pivot bearing": W. J. Harrison, M.A., *Trans. of the Cambridge Phil. Soc.*, Vol. xxii, No. XVIII, p. 373–388.

CHAPTER IV

MECHANICAL TESTING OF LUBRICANTS

THE mechanical testing of a lubricant should be effected in a machine which reproduces, as closely as possible, the conditions under which the lubricant is subsequently required to work. In this connection, particular care should be paid to the speed, load, temperature, and method of applying the lubricant, but other conditions, which may produce an important effect on the results obtained, may need to be represented.

When the oil is required for lubrication at low speeds where the oiliness or greasiness is the most important factor, the testing machine should run at corresponding surface speeds and loads in order that the viscosity effect may be reduced to a minimum value.

If, on the other hand, the lubricant is to be used under conditions of complete or perfect lubrication at comparatively high speeds, the losses due to internal friction or the viscosity effect are the important factor to determine.

Some types of testing machines are adaptable for determining the difference due to different combinations of bearing surfaces and, as this effect can only be demonstrated when there is partial metallic contact, a low speed is necessary in order to ensure that the effect of viscosity is reduced to a minimum.

Oil-testing Machines. One of the most common types of testing machine in general use is that in which the lubricant is introduced between a loaded bearing and a shaft. The latter is rotated and provision is made whereby the frictional moment on the bearing is measured. This class of machine, when run at moderate speed, is, however, only useful in determining the relative frictional resistances due to the viscosity of the lubricants, and comparisons of this effect can almost equally well be made by determinations of their absolute or comparative viscosities. It is claimed for many machines of this type that it is possible to determine accurately the coefficient of frictional resistance, but this claim cannot always be substantiated, for in some designs there are errors inherent in the construction which render this measurement liable to very considerable inaccuracy.

Thurston Oil-testing Machine. A machine of good design and one which is in very general use is the Thurston oil-testing machine, manufactured by Messrs. Sir W. H. Bailey & Co., Ltd., and illustrated in Fig. 7. This machine consists of a shaft, *A*, mounted in two bearings attached to the main supporting frame, *B*. At one end, the shaft is overhung and carries the test-bearing, *C*, which consists of two half-bushes mounted on opposite sides of the shaft. These bushes are supported in position by the housing, *D*, to which the loading spring is attached. The brasses, with their housing, spring and spring case, form a compound pendulum, the inclination to the vertical of which gives a measure of the frictional resistance between the bearing and the shaft. The loading arrangement can be seen clearly in Fig. 7. A tube, *E*, screwed into the underside of the housing contains the loading spring, and the closed lower end of the tube forms the abutment for the spring. The upper end of the spring presses, through a suitable plate, on the underside of the lower half-bearing, which is thus forced against the shaft, being resisted through the housing by the half-bush on the top side pressing downwards against the shaft. Thus the load on each half-bush is measured by the compression of the spring, and this can be adjusted by rotating the central spindle, *F*, carrying a collar, *G*. The amount of compression in pounds is indicated by the scale on the tubular spring-container by a pointer connected with the collar, *G*. The load can readily be released by rotating the nut, *H*, until it takes the thrust of the spring, the load then being reduced to the weight of the pendulum. The overhanging portion of the shaft is shrouded by a sleeve, *K*, which forms the true journal, and this arrangement enables various materials to be used for the revolving surface.

In order that the bearing shall wear evenly over its whole length, and to assist in spreading the lubricant, a device is fitted which gives the bearing and its attachment a small reciprocating lateral motion. The housing of the bearing receives its motion from a small spindle, *L*, coaxial with and passing through the shaft, the latter being formed hollow to receive it. The oscillating motion is given to the spindle by a small eccentric, *M*, driven through a worm reduction gear, the worm being mounted on the shaft and the worm-wheel on the eccentric spindle.

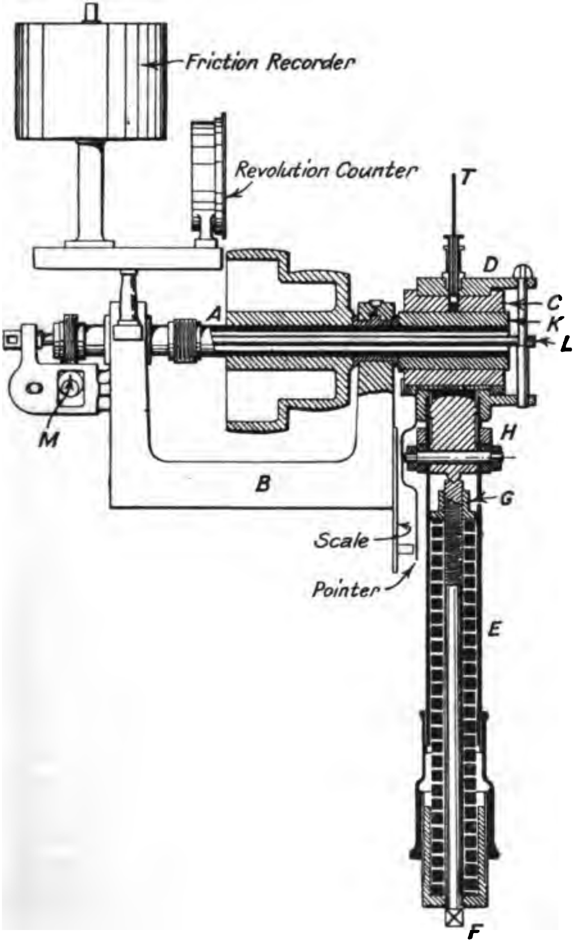

FIG. 7.—THURSTON'S OIL TESTING MACHINE.

A thermometer, *T*, the bulb of which is immersed in a tube containing mercury, is provided as shown, in order to ascertain the temperature of the lubricant. The lower end of the thermometer tube passes through a hole provided in the upper bearing bush and is separated from the journal by only a very small distance, so that the mercury and, therefore, the thermometer, readily follows the variation in temperature of the rubbing surfaces.

Provision is made whereby the number of revolutions of the shaft and the frictional resistance are recorded automatically.

A belt pulley is provided on the shaft for running the machine, but it is more satisfactory, especially at low speed, to employ an electric motor and drive direct or through a flexible coupling, as the pendulum is apt to give unsteady readings if either the joint in the belt or the material of which the belt is made, is unsuitable.

Each bearing block subtends an angle of about 60° or 70° at the centre of the shaft. The spaces then left at the sides, where the shaft is exposed, are utilized for applying the lubricant, and the method selected for doing this can be varied from feed lubrication to bath lubrication. Mr. Archbutt feeds in the oil by means of tubes, each of which is drawn down at the end to a fine orifice through which is threaded a cotton wick, the latter being held in position by non-corrosive wire. The cotton wick is allowed to wipe against the journal, which it wets with oil. (*See* Fig. 8.) A needle, rolling on the shaft and held in position close up to the "on" side of the block, serves to spread the oil over the top of the journal. On the lower block a small metal strip attached to the "on" side performs a similar function.

The pressure on the bearing is calculated on the projected area of contact of the blocks. If both brasses have the same area of contact, the pressure intensities differ, as the upper brass supports the weight of the pendulum in addition to resisting the load of the spring; this difference, however, is only of importance when the load is small.

The frictional resistance between the bearings and the shaft causes the pendulum as a whole to swing out of the vertical, and this deflection is indicated on a scale, a pointer being attached to the pendulum.

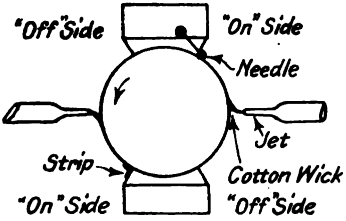

FIG. 8.—METHOD OF APPLYING LUBRICANT IN THURSTON'S MACHINE.

The indication of the pendulum for a given frictional force depends upon its weight and upon the distance of its centre of gravity from the axis of the shaft. When, however, the spring is adjusted, the radius of the centre of gravity of the pendulum is altered by a small amount, and the scale of frictional resistance cannot, therefore, be true for all positions of the spring unless an adjustable compensating weight is provided, which is usually the case.

Determination of Coefficient of Friction by Thurston's Machine. In general, the coefficient of friction is given by

$$\mu = \frac{F}{L}$$

where F = total frictional resistance indicated on the pendulum scale,
and L = load as indicated on the spring tube.

For more accurate work, it is better to graduate the pendulum scale in degrees of arc, and to determine the coefficient of friction by calculation from first principles.

Thus, if W = weight of the pendulum,
F = total frictional force,
θ = inclination of the pendulum to the vertical,
R = radius of the journal,
R_1 = radius of the centre of gravity of the pendulum,

then $FR = WR_1 \sin\theta$

and $$F = \frac{WR_1 \sin\theta}{R}$$

Hence $$\mu = \frac{WR_1 \sin\theta}{LR}$$

Notes on Use of Thurston Machine. In order that the surfaces of the journal and bearing blocks shall not be damaged, it is necessary to apply the load only when the machine is running, and to release it before stopping.

The greatest care must be taken with the rubbing surfaces. The brasses must be bedded-in carefully, until contact takes place over the whole of the bearing surfaces ; otherwise heat would be developed locally and seizing would ensue. The preparation of the brasses takes considerable time and a certain amount of skill and, when once they are properly bedded and present a good surface to the journal, care must be taken to keep them so if good comparative tests are to be made. It is useless to continue testing if seizing once takes place, even though it may be slight, and it must be continually borne in mind that the frictional resistance obtained depends considerably on the condition of the rubbing surfaces.

Mr. Archbutt has successfully adapted a large Thurston machine for running at very low speeds and under heavy loads. Under these conditions, results have been obtained which are quite independent of the viscosity of the lubricants ; such results may be considered much more valuable, as they indicate the true lubricating value of the lubricant and are not merely a measure of its internal frictional resistance.

Goodman's Testing Machine for Bearing Metals and Oils. In this machine, a single-bearing brass, having a small arc of contact with the journal, is loaded through a lever system in such a manner that the brass and its housing are free to execute small oscillations about the centre of the shaft. Fig. 9 shows diagrammatically the arrangement of the machine. The shaft is supported by very large friction discs, *W*, with bearings of ample dimensions. The employment of these discs eliminates the heat which is developed in ordinary bearings and which, by conduction along the shaft and by radiation, interferes somewhat with accurate measurements of temperature of the test bearing.

Load is applied to the bearing by a jockey weight, *C*, acting through a double lever system.

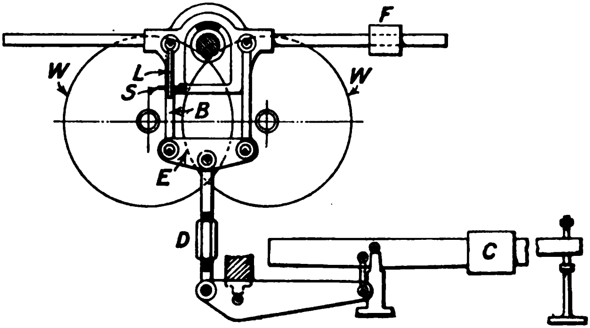

Fig. 9.—Diagram of Goodman's Oil Testing Machine.

The load is transmitted from the latter to the bearing housing by means of a turnbuckle, *D*, and a linkage, *E*. The turnbuckle serves to apply and release the load.

The lever system is fitted with knife edges throughout, and it is essential that the knife edges and the plates on which they rest should perform their functions perfectly.

The frictional torque on the bearing housing tends to revolve it in the direction of rotation of the shaft, but its symmetrical position is maintained by adding or moving the small jockey weight, *F*, until the balance is re-established.

Both the load beams and the arm attached to the bearing housing are prevented from moving too far in either direction by suitable stops.

Normally the turnbuckle link should be attached to the beam midway between the points of attachment of the two vertical links. Any want of symmetry in this respect can be balanced at the measuring arms. If, however, the knife edges become damaged, the load may cease to act through a point which remains at a constant distance from the two link knife edges, and in this case the balance of the bearing housing is in error by an amount equal to the whole bearing load multiplied by the shift of the point of its application on the centre knife edge. If the load is great and the frictional resistance of the bearing small, the error produced in this manner due to the load shifting on the knife edge, after the zero has been taken, may effectually swamp the frictional measurement and render the results obtained useless.

This error is inherent in this method of applying the load, but can be avoided in a considerable degree if care is taken that the knife edges are not damaged.

Professor Goodman has obtained accurate initial adjustment of the main knife edges by the arrangement shown in Fig. 9. One of the knife edges is fixed to a lever, *L*, and by means of the screwed rod and locknuts shown at *S*, the knife edge can be slightly rotated in the end of the link, *B*, until its distance from the shaft centre is exactly equal to the distance of the centre of the shaft from the opposite knife edge.

This machine is manufactured by Messrs.

Riehlé Bros., of Philadelphia, U.S.A., and is used as the U.S. Standard Machine. Fig. 10 shows the general view of the machine.

Deeley Oil-testing Machine. This machine has been designed primarily for the study of the oiliness of lubricants and the variations in frictional resistance produced by different combinations of metallic surfaces.

It consists essentially of a rotating dish, above which is supported a device for measuring and indicating torque. Of the two frictional surfaces to be experimented upon, one is in the form of a disc which rests on a revolving dish, by which it is rotated ; and the other takes the form of three small pegs projecting from the underside of a light carriage, to which they are firmly attached. The carriage is centred by a spindle on the axis of rotation of the disc, and is provided with two vertical arms, symmetrically placed about the centre, which engage in the torque measuring device above. The lubricant is placed in the dish so as to cover the experimental plate, and the carriage is placed in position, the opposing surfaces being separated by a thin film of lubricant. When the dish is rotated, the frictional drag between the opposing surfaces causes the carriage holding the pegs to be rotated also ; but, in doing this the torque-measuring apparatus is engaged, and a gradually increasing torque is applied to the carriage. In due time, slip will occur, and the torque necessary to produce this slip is indicated. From the torque measured at the moment slip occurs, the static friction is calculated.

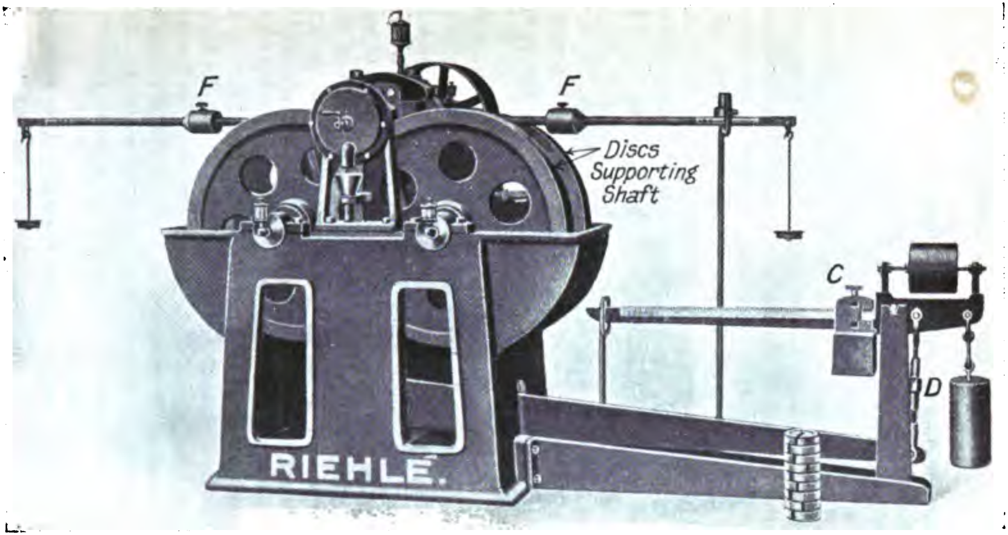

FIG. 10.—RIEHLÉ (GOODMAN'S) OIL TESTING MACHINE.

The machine is shown in Figs. 11 and 12. The dish *b* (Fig. 11) is driven through bevel gearing as shown, the pinion being attached to the shaft, *g*, which may be rotated either by hand with a small handle provided or by a small motor through the agency of reduction gearing. The lower experimental surface is shown at *A* (Fig. 12), and the pegs forming the opposing surface are shown at *B* (Fig. 12). The frame or carriage, *C* (Fig. 11), to which the pegs are attached, is centred by the spindle, *S*, three rollers held by the screws *R* (Fig. 12) forming a roller-bearing on the carriage.

For varying the loads, equal weights, *w*, are provided, and each corresponds to an intensity of load of 10 lbs. per square inch when the pegs, *B*, are $\frac{5}{32}$ in. diameter.

The vertical posts of the carriage are fitted with rollers which take a bearing on the arms of the crosshead *p*. This crosshead is fixed to a vertical spindle, *a*, running in roller bearings. To the spindle is attached the centre of the spring, *O*, the outer end of which is clamped to the frame of the machine. A train of wheels is provided on the torque spindle with the object of damping oscillations due to rapid changes of the friction between the test surfaces ; that wheel, which rotates at the greatest speed relative to the others, is made with a heavy rim in order to increase its inertia and so assist further in damping. Two pointers are provided, one being fixed directly to the spindle and one on an auxiliary spindle rotating at ten times the speed.

When slip takes place between the opposed surfaces, a thick oil film is formed between them,

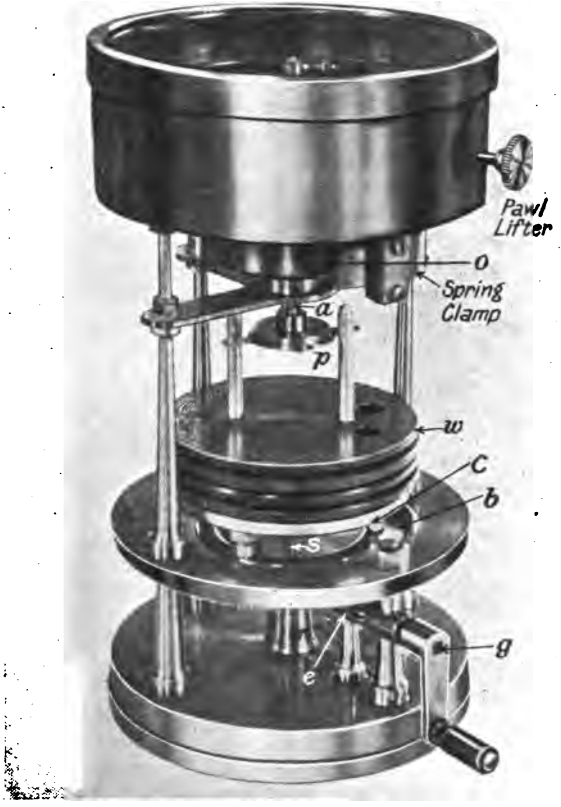

FIG. 11.—DEELEY FRICTION MACHINE.

the frictional resistance is reduced very greatly, and the spring in giving out its stored energy tends to rotate the carriage at a high speed in the opposite direction to that in which it was previously moving. To prevent this action, in order to allow the maximum reading of the pointer to be taken, a ratchet is provided on one of the wheels in the train; and a pawl, which can be raised at will, engages in the ratchet.

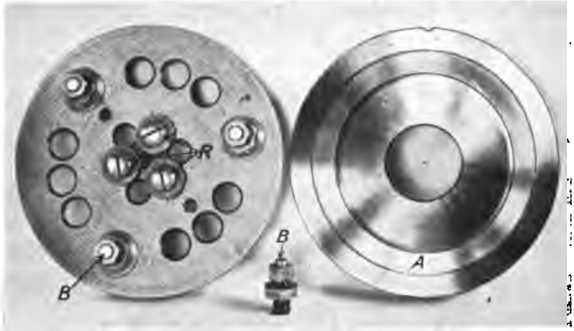

FIG. 12.—EXPERIMENTAL SURFACES FOR DEELEY MACHINE.

The greatest care has been taken in the design of the machine to reduce the friction of the torque-measuring apparatus. The calibration of the torque spring is effected by replacing the crosshead by a pulley and applying a known torque to the spindle. The readings obtained on the scale are found to be exactly proportional to the torque; and a scale of friction could be substituted for that marked on the apparatus, in which one turn of the pointer corresponds to one hundred divisions, if the zero of the spring is adjusted carefully.

The preparation of the experimental surfaces presents considerable difficulty and, in order to obtain exact results, a fair amount of experience with the machine is necessary. The surfaces must be ground under water and every trace of abrasive removed. At the same time, both disc and pegs must be kept perfectly flat, and every endeavour must be made to produce in comparative experiments the same degree of polish, and this, as experience shows, must be a very high polish. The rubbing surfaces, after having been prepared in this way, will readily seize together and should not be allowed to slide over each other under load. Once seizing has taken place, however slightly, the friction will be increased considerably. Seizing is indicated by scratches on the faces of the pins and on the disc.

This machine is at present in a more or less experimental stage, and the difficulties experienced in preparing the friction surfaces will undoubtedly be reduced considerably. Recent experiments on various lubricants, and with different combinations of metallic surfaces, go to show that very reliable and useful information on the relative values of lubricants can be obtained by its use.*

The Daimler-Lanchester Worm Gear-testing Machine. This machine has been used at the National Physical Laboratory with some success as a means of measuring the differences due to lubricants under conditions of extremely high load. The machine was originally designed by Dr. F. W. Lanchester, F.R.S., for the determination of efficiencies of worm gears at the Coventry Works of the Daimler Co. It was later presented to the National Physical Laboratory, where it is now installed.

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* For results of tests carried out by the author on the Deeley Machine see *Engineering*, June, 1920.

It is proposed here to give a short description of this ingenious machine and to discuss in Chapter VII some results which have been obtained by its use in testing lubricants.

A diagram of the apparatus is shown in Fig. 13. The worm gear is contained in the box, *A*, which is held in a special frame, *BB*, consisting of two large friction discs, *BB*, joined by bridges B_1B_1. The gear box proper is provided with trunnions, on an axis parallel to that of the worm, which are carried in ball bearings mounted in the bridge pieces. The friction discs are supported on four small ball bearing rollers, enabling the gear box and the frame to rotate about an axis parallel to the axis of the worm-wheel. Thus supported, the worm-wheel can execute small oscillations about two axes parallel to the worm and worm-wheel axes respectively.

When power is transmitted through the gear, the reaction torques on the gear box tend to make it to rotate, but this tendency is checked by a load, *P*, hung from the knife edge, *K*. It will be seen from the sketch that the load hangs from a bracket fixed at right angles to the end of the torque arm, *G*, which is attached rigidly to the gear box. The load thus supported produces a moment about both axes of rotation. Consideration will show that, by varying the horizontal distance of the knife edge, *K*, from the axis of

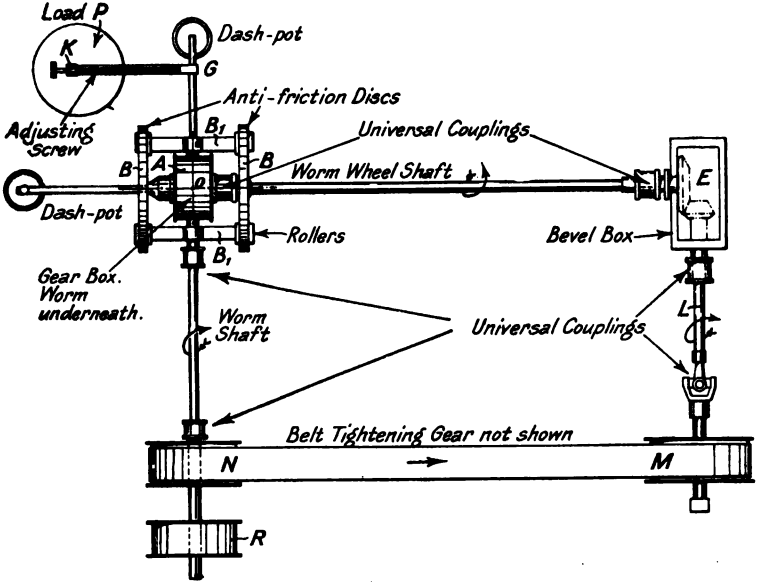

Fig. 13.—Daimler-Lanchester Worm Gear Testing Machine Installed at the National Physical Laboratory, Teddington.

the worm, the gear box may be balanced for any load.

If we assume that the efficiency of the gear box is 100 per cent and the gear ratio, say, three to one, then the torque about the axis of the worm will be one-third of that about the axis of the worm-wheel. The balancing load being the same for each torque, the distance, *KG*, will be one-third of the distance *GO*, i.e. if *GO* is 24 in., then *KG* will be 8 in.

In practice, of course, the efficiency of the gear is less than 100 per cent, and the knife edge carrying the load on the bracket, *GK*, will need to be shifted from the 8-in. mark until balance of the gear box is restored. The movement of the load with respect to the worm axis is controlled by a micrometer screw gear.

The efficiency of the gear is obtained from the ratio of the theoretical distance of the load from the worm axis (assuming 100 per cent efficiency) to the actual distance measured. Thus, if balance is obtained when the distance, *KG*, is, say, 8·55 in., then the efficiency = 8·00/8·55 = 93·5 per cent.

The worm is driven from the pulley, *R*, through a cardan shaft provided with special ball-bearing universal joints. The drive is taken from the worm-wheel through a similar shaft to the bevel box, *E*, where it is again turned through a right angle. The ratio of the bevel gear is of the same order as that of the worm gear under test, but the wheel is made the driving member and the shaft, *L*, therefore, runs at approximately the same speed as the worm shaft. The shaft, *L*, is provided at each end with universal couplings, one of which allows the shaft considerable lateral motion. The pulley, *M*, is mounted in a vertical swinging frame pivoted some 2 ft. below the shaft axis, and a screw gear is provided which controls the angle that this frame makes with the vertical and, therefore, the distance apart of the pulleys *M* and *N*. By this means the tension of the belt can be adjusted to and maintained at any desired value, and the power transmitted through the worm gear can be varied within fine limits. The belting arrangement not only provides a means of adjusting the load; the pulleys, *M* and *N*, are selected of such diameters that the peripheral speed of *M* is slightly in excess of that of *N*; and, when this is the case, the power transmitted round the circuit, consisting of the worm shaft, worm gear, wheel shaft, bevel gear, and belt, may be as much as ten times the power required to drive the machine from an external source.

A little consideration will show how this object is achieved. When the belt is slack, the power transmitted through the worm gear is determined by the resistance to motion of the worm-wheel shaft, the bevel gear and the pulley, *M*, together with the resistance of the various bearings supporting them.

When the belt is tight, assume, firstly, that it is so tight that there can be no slip over the pulleys, *M* and *N*; and, secondly, that the worm shaft and the shaft, *L*, are constrained by the worm and bevel gearing to run at the same speed, and that the pulley *M* is slightly greater in diameter than the pulley, *N*. Under these conditions, if the worm shaft is turned, the first part of the initial revolution will take up all the back lash in the gears and then one of two things must happen: either the belt must slip, as it cannot run over pulleys constrained to move at different peripheral speeds, or the worm teeth must shear. (It is assumed, of course, that the worm gear is the weakest unit in the machine.)

We have, therefore, the two extremes; the one, when the belt is quite slack, of extremely little power transmitted through the gear, when running at any speed; and the other, of load sufficient to shear off the wheel teeth. Obviously between these two limits the belt can be adjusted to correspond with the transmission of any power through the worm gear, within the limits for which the machine is designed. It will be noted that the power required to drive the machine is only the power lost in the gear boxes, bearings, and at the surfaces of the pulleys due to the belt friction. The machine is driven by a 15 h.p. petrol motor which provides sufficient power to drive the machine when the pressure between the gear teeth corresponds to the transmission through the gear of as much as 150 h.p.

In making a test of efficiency, weights are hung from the knife edge to produce the required torque, and the machine is run up to speed. The belt is then adjusted until the weight is lifted, and the gear-box is balanced about the worm-wheel axis. Adjustment of the knife edge position is then made by means of the screw gear provided, until the balance is obtained about the worm axis and the position for balance is noted.

Suitable stops are provided to confine the movements of the gear box within reasonable limits, and heavy dashpots to prevent violent oscillations due to variations in belt grip and other causes.

The machine requires to be set up accurately, in order to ensure that the angle between the shafts leading to the worm and from the worm wheel shall be exactly at right angles.

For loads up to from 60 to 80 h.p. at 1,500 r.p.m., an absolute accuracy of ± 0·2 per cent on efficiency is obtained, and differences in efficiencies can be determined to within one-half of this amount.

When used for testing oils, the lubricant is contained in a tank, from which it is pumped up through a flexible pipe to the top of the gear box and run in over the worm-wheel. A free outlet is provided near the bottom of the box, and the excess oil returns to the tank. Circulating in this manner provides a means of regulating the temperature of the lubricant, which is very necessary when the efficiency varies considerably as the oil becomes hot.

Iron-constantan thermocouples are fitted at both inlet and outlet, but at ordinary speeds there is no appreciable difference in the temperatures recorded.

Fig. 14 shows the arrangement of the gear box. The screw gear for shifting the load is not shown, but is fixed to the bracket, *X*, and shifts the load backwards and forwards at right angles to the surface of the paper.

The ball bearings supporting the trunnions on the gear box and the friction discs will be noted, and it will also be seen that these allow for oscillations in directions at right angles.

Recent work on lubrication, which has been carried out on this machine for the Department of Scientific and Industrial Research, is described

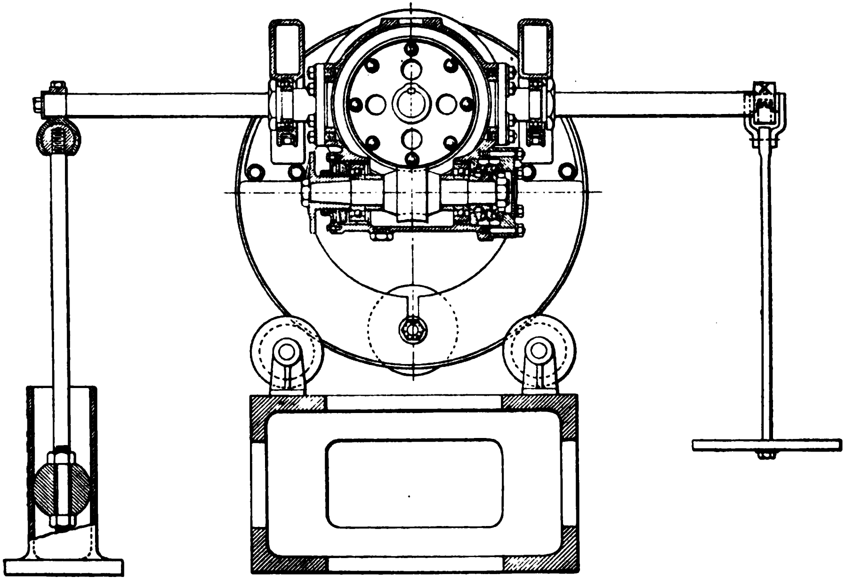

FIG. 14.—GENERAL ARRANGEMENT OF WORM CASING IN DAIMLER-LANCHESTER MACHINE.

in Chapter VII. There can be no doubt that the machine truly represents conditions of lubrication in which the pressure on the lubricating film is of very great intensity, and that the results of experiment which have been made are applicable to most forms of gearing.

CHAPTER V

PHYSICAL AND CHEMICAL TESTING OF LUBRICANTS

Measurement of Viscosity. The most accurate and reliable method of determining the viscosity of a liquid is that employed by Poiseuille, and by Thorpe & Rodger, which consists in determining the rate of flow of the liquid through a capillary tube under known conditions of pressure and temperature. An absolute viscometer of this type has been employed by Mr. W. F. Higgins, of the Physics Department of the National Physical Laboratory,* and reference to his work should be made for a complete description of the apparatus and some results obtained by its use.

The apparatus employed is shown in Fig. 15, and consists essentially of a U-tube, the vertical limbs of which are about 2-cm. bore, connected at their lower ends by a capillary tube of known dimensions. The U-tube is completely immersed in a water-bath provided with an electric-heating coil and a stirrer, and by regulation of the current through the heater the temperature can be maintained at a constant value within 0·2° C. for a considerable period. The vertical limbs of the U-tube are etched with horizontal lines near the top and bottom, and the volumes between the upper and lower marks are determined accurately.

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* "On the Methods and Apparatus used in Petroleum Testing," Part II—Viscometry, by W. F. Higgins, A.R.C.Sc., B.Sc.: *Petroleum World,* July, 1913.

The viscometer is filled with liquid up to the level of marks placed centrally between the upper and lower marks on the vertical tubes. In carrying out an experiment, it is arranged, in the first place, that the liquid shall stand level with the upper mark on one tube and, therefore, with the lower mark on the other. The tube in which the liquid is at the high level is next placed into communication with a large reservoir containing air under pressure, and the liquid is driven down the tube and through the capillary, rising in the adjacent tube. The time of transit of the meniscus

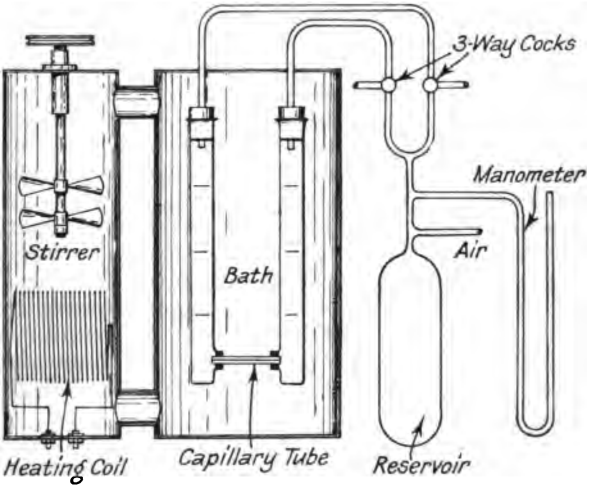

FIG. 15.—ABSOLUTE VISCOMETER.

from the upper to the lower mark is observed by means of a stop-watch. It will be observed that the pressure causing the flow is made up of the pressure due to the air and the difference in head of the liquid in the two limbs of the U-tube. Consideration will show, however, that if the tubes are of equal bore and the distances between the marks in the two tubes are the same, the pressure due to the liquid head will assist the flow while the liquid is falling from the upper level to the central mark, and resist the flow when the liquid falls from the central mark to the lower mark, and, therefore, its average value will be zero. The air pressure is maintained practically constant by the employment of a large air reservoir, and its intensity is determined by a manometer. Certain corrections have to be made for inequalities in the volumes between the marks of the two tubes, as it is difficult in practice to obtain tubes of precisely the same diameters.

The absolute viscosity is calculated from the pressure and the rate of flow as described in Chapter I.

Commercial Viscometers. The absolute viscometer is not used extensively in the oil trade. It is replaced by an instrument in which a certain quantity of oil, at a known temperature, is allowed to flow through an orifice or short tube, the time of flow being measured. The time of outflow is compared with that obtained under the same conditions for a standard liquid with the same instrument, and the number expressing the ratio of the times of outflow is termed the "viscosity." Very frequently the "viscosity" is expressed simply as the time of outflow and in seconds. It must be borne in mind, however, that the value of the viscosity expressed either as a time or as a ratio is dependent upon the particular instrument in which the tests are made. Absolute viscosity is, as shown in Chapter II, a shear stress, and should be expressed in shear units.

Of three types of viscometers in common use, the Redwood is used in Great Britain, the Engler in Germany, and the Saybolt in America. A diagram of the Redwood instrument is given in Fig. 16.

In all three types, the oil is contained in a cup provided with an indicator or mark for the initial level of the oil. The cup is supported in a bath containing oil or water, the temperature of which can conveniently be regulated. When the oil is at the required temperature, it is allowed to flow out through the orifice into a graduated flask beneath the instrument, and the time taken for the outflow of a standard volume is measured by a stop-watch.

Redwood Viscometer. In using the Redwood instrument (Fig. 16), the following precautions must be observed if accurately comparative results are to be obtained.

In the first place, the oil cup must be perfectly clean. It should be washed out with benzine and afterwards with ether. The agate jet should be cleaned with the same solvents and finally scoured inside with a twist of tissue paper. The greatest care must be observed in preserving the sharp edges to the ends of the orifice, as the slightest chipping of these will ruin the jet. The

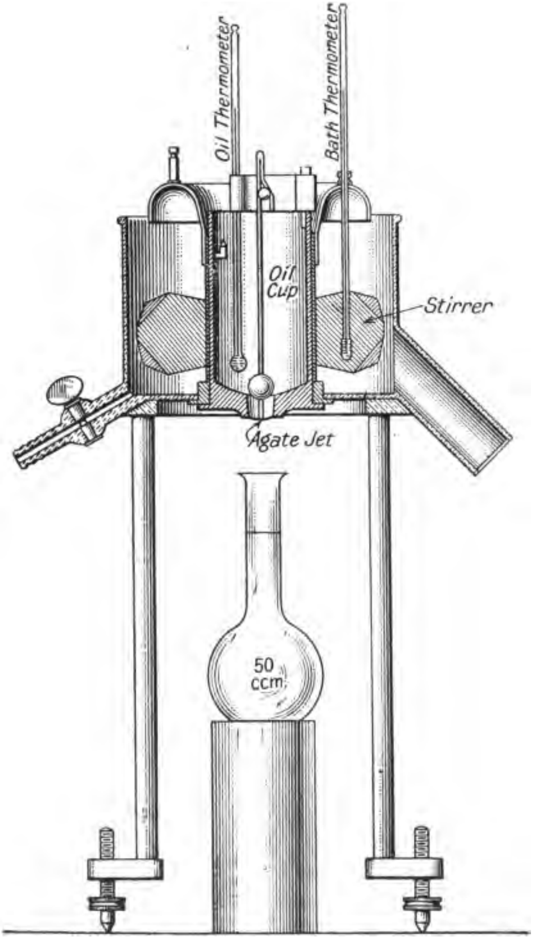

FIG. 16.—REDWOOD VISCOMETER (NO. 1).

apparatus must be levelled by means of the screws and level provided for this purpose.

The water bath should be filled with water before filling the oil cup, in order to avoid splashing water into the oil cup, which is very likely to occur if the reverse process is adopted. It is best to fill the oil cup a little above the filling mark and run off the superfluous oil just before commencing a test, in order to ensure that the jet should contain oil at the same temperature as that in the cup. This operation should be repeated at each temperature at which tests are made.

The flask used for receiving the outflowing oil should be placed quite close to the bottom of the jet to guard against a draught of air blowing the stream of oil against the neck of the flask and so obscuring the meniscus as it passes the graduation on the flask.

In raising the temperature of the bath, it is advisable to take it a little above the temperature required, in order to raise the temperature of the oil more quickly. At the same time, the thermometer dipping into the oil cup may be used as a stirrer.

The bulb of the oil cup thermometer should be placed sufficiently low in the cup to enable it to record the true temperature when the desired amount of oil has been run out, but care should be taken that it is not placed low enough to impede the flow through the orifice.

Mr. Higgins has found that, in order to obtain results to an accuracy of 1 per cent, it is necessary to set the initial level of the oil to within 0·7 mm. of the filling mark.

Formulae for Commercial Viscometers. The Redwood viscometer number is obtained thus: "The number of seconds occupied by the outflow of 50 cu. cm. of the oil is multiplied by 100 and divided by 535, the average number of seconds occupied by the outflow of 50 cu. cm. of rape oil at 60° F. from the apparatus. The result is multiplied by the specific gravity of the oil at the temperature of testing and divided by 0·915, the specific gravity of refined rape oil at 60° F." * The number obtained represents the "viscosity" in terms of rape oil under the same conditions.

If η be the absolute viscosity of the liquid under test, T the time of outflow of 50 cu. cm., and ρ the specific gravity of the liquid, then the value of η in C.G.S. units may be determined by the following equations—

For the Redwood Viscometer—

$$\eta = \rho\left(0{\cdot}0026T - \frac{1{\cdot}715}{T}\right)$$

For the Engler Viscometer—

$$\eta = \rho\left(0{\cdot}001435T - \frac{3{\cdot}22}{T}\right)$$

For the Saybolt Viscometer—

$$\eta = \rho\left(0{\cdot}00213T - \frac{1{\cdot}535}{T}\right)$$

In Fig. 3 (p. 12) is shown the variation of viscosity with temperature for various oils of animal, vegetable, and mineral origin.

Fig. 17 shows the value of Viscosity/Density

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* See *Petroleum and Its Products*, by Sir Boverton Redwood. (Vol. ii.)

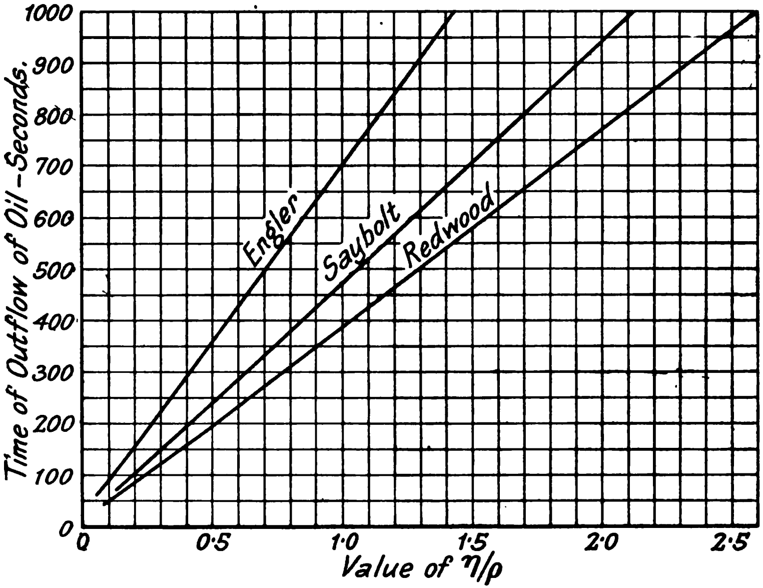

FIG. 17.—VALUES OF VISCOSITY : DENSITY RATIO (η/ρ) FOR OUTFLOW TIMES UP TO 1,000 SECS. FOR ENGLER, SAYBOLT AND REDWOOD VISCOMETERS.

(η/ρ) for times of outflow up to 1,000 secs. for the three instruments, Redwood, Engler, and Saybolt. The dimensions of orifice and the volume of outflowing oil in each instrument are as shown in Table I.

TABLE I

DIMENSIONS OF ORIFICE AND VOLUME OF OIL FOR COMMERCIAL VISCOMETERS

Viscometer.	Orifice: Length × Diam.	Volume of Outflowing Oil.
	mm.	cu. cm.
Engler . .	20 × 2·8 to 2·9	200
Saybolt . .	13 × 1·8	60
Redwood. .	10 × 1·5	50

Specific Gravity. The density of a substance is defined as its mass per unit of volume.

The ratio of the mass of any volume to the mass of an equal volume of water at the same temperature is called the specific gravity.

Thus, density is expressed in units of mass (pounds or grammes) per unit volume (cubic inches or cubic centimetres), but specific gravity is simply a number.

Determinations of specific gravity may be made by the use of the specific gravity bottle, or with a hydrometer or by means of the specific gravity balance. It is not proposed here to give a detailed description of the various methods employed; for these the student is referred to text-books on Physics, and to the work of Messrs Archbutt & Deeley on the subject.*

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* *Lubrication and Lubricants*, by Archbutt and Deeley (Charles Griffin).

The variation in specific gravity with temperature is important in the determination of absolute viscosity, and typical examples of this variation for temperatures up to 100° C. are given in Fig. 18.

Cold Test. The "cold" or "pour" test of an oil consists in a determination of the temperature at which the oil changes from the liquid to the solid state. As the temperature is reduced, most of the fixed oils and some of the mineral oils deposit solid fats or waxes; but in the case of Russian oils, in which no definite solids are deposited, the oil becomes more and more viscous, and forms a soft jelly which will not flow.

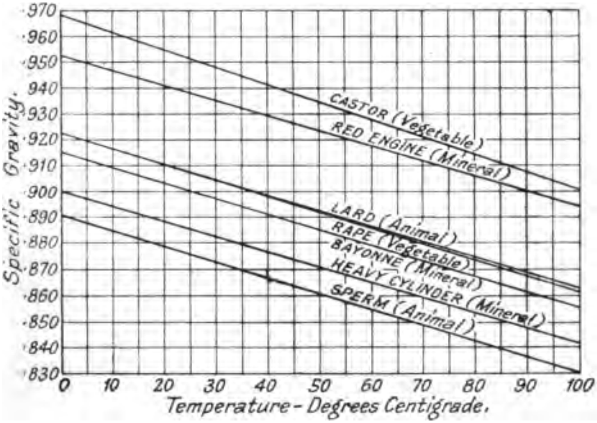

FIG. 18.—VARIATION OF SPECIFIC GRAVITY WITH TEMPERATURE.

The method of test employed consists of placing in a freezing mixture a test tube containing the oil to be examined, and observing the temperature at which the oil ceases to flow. This temperature may be observed *either* by gradually reducing the temperature and noting the thermometer reading when the oil begins to set, *or* by initially cooling the oil until it is well set, and then removing it from the freezing mixture and noting the temperature at which the last trace of solid disappears. If there is no considerable difference between the setting and melting temperatures thus determined, the mean of the two readings may be taken. In any case, the method adopted should be specified in order to avoid confusion.

Emulsification. In cases where it is not possible to prevent the access of water to a lubricant, as in turbine gearing and in steam cylinder lubrication, it is essential that the oil and water shall readily separate after having been shaken vigorously together. Some mineral oils when treated in this manner, especially if a trace of lime be present, form a thick emulsion which ultimately becomes a "tacky" mass, totally unfit for lubrication.

Turbine gears used in ships cannot be cleaned out at frequent intervals, and partial emulsification of the lubricant not only means inefficient lubrication, but loss of a large quantity of valuable oil.

In cases of steam cylinder lubrication, if the oil emulsifies, it is not deposited in the oil separator, but is carried away by the exhaust steam in infinitesimally small emulsified globules and deposited in the condenser or carried on to the boilers. As, however, emulsification of cylinder oil nearly always exists in a slight degree, specially designed apparatus is used for extracting both the separated and emulsified oil from the feed water.

Flash Point—Open Test. Determinations of the "open" flash point can readily be made by using a small porcelain crucible and a sand bath. The crucible should be about 2 in. diameter and 1 in. deep, and the sand bath should be deep enough to allow the crucible to be embedded, so that the lip just appears above the surface of the sand; silver sand should be used. The bath is supported on a tripod and is heated from below by means of a Bunsen burner. A thermometer should be supported vertically in such a manner that its bulb is central with and about ¼ in. above the bottom of the crucible. The oil under test should reach to within about ¼ in. of the crucible top. The test flame is best made up from a piece of small bore brass tubing, connected to a gas regulator by means of which the size of flame is regulated.

In carrying out an experiment, the flame is passed across the surface of the crucible every time the temperature of the oil has risen a few degrees. It is essential that the flame should not touch the oil, although it should pass very near its surface. When a flash occurs, a small bluish flame will pass over the whole surface of the oil. At the first flash the temperature is noted; this is the "open" flash point. If the heating is continued, a temperature will be reached at which the oil continues to burn, and this temperature is defined as the "firing" point.

Values obtained by this method are only

approximate, and are always considerably higher than those obtained by using a standard flash-point apparatus.

Flash Point—Closed Test. Two types of instrument are in common use for this test, namely, the Gray and Pensky-Martens, but the latter is the better designed apparatus, and is more convenient to handle at high temperatures. The Pensky-Martens apparatus consists of a brass cup fitting loosely in a cast-iron vessel, by which it is supported in such a manner as not to touch on either the sides or the bottom. The cup is closed at the top with a cover fitted with a perforated slide, and a spring lever is provided which, when turned, rotates the slide until its perforations correspond with similar perforations in the cover. The test flame is caused to dip through a central opening into the air space above the oil in the cup. A double stirrer is provided, with vanes in both the oil and in the air space above it. A thermometer dipping into the centre of the body of the oil indicates its temperature. A by-pass is fitted over the test flame, which avoids the necessity of continually relighting the gas jet.

In order to obtain consistent results, the following points must be observed—

1. The positions of the stirrers must be adjusted so that they properly perform their functions, one of stirring the oil, and the other of stirring the mixture of oil vapour and air above the oil.

2. The size of the test flame must be kept constant.

3. The rate of heating the oil must not be too rapid, and the heater should be regulated so

that the rate of rise in temperature, when near the flashpoint, may be reduced.

4. The stirrers should be used before applying the test flame.

5. In using the apparatus, it is best to make a preliminary experiment in order to ascertain the flash point of the oil under test within a few degrees.

Constitution of Lubricating Oils. In order to ascertain the purity, composition and suitability of oils to be used as lubricants, it is necessary to carry out a chemical examination, in addition to the physical tests (i.e. viscosity, flash point, etc.).

Before describing the chemical examination, it may be as well to indicate the chemical constitution of oils in general use as lubricants. For this purpose, the oils can be divided into two main classes; (1) Mineral oils and greases; (2) vegetable and animal oils and fats. The mineral oils and greases consist generally of a more or less complex mixture of hydrocarbons which do not contain oxygen, and are stable chemical compounds, not easily attacked or decomposed by chemical reagents. The vegetable and animal oils, on the other hand, consist usually of mixtures of what are known as glycerides—stearin, palmitin, and olein being the best known. Most of the fatty oils are composed of a mixture or combination of these three glycerides in varying proportions together with, in the case of individual oils, proportions of similar and more complex compounds; for instance, rape oils contain the glycerides of rapic and erucic acids.

One important exception to the general rule is castor oil, consisting mainly of a glyceride known as triricinolein, which accounts for some remarkable differences between its properties and those shown by other fixed oils.

Unlike the mineral oils, the fatty oils contain oxygen, and are susceptible in varying degree to the action of chemical reagents. Thus they can be hydrolyzed, or split up, with the formation of glycerol and the liberation of fatty acids, which latter combine with alkali to form soaps. Advantage is taken of this fact in the determination of the "saponification" value described later.

The fatty oils can be divided roughly into two classes: (*a*) those consisting of saturated compounds; (*b*) those containing a greater or less proportion of unsaturated or volatile compounds. To the latter belong the drying oils, which, owing to the presence of these unsaturated compounds, combine readily with oxygen, free iodine, etc. The action of iodine is used in the determination of the "iodine value."

Chemical Tests Applied. The chemical examination which is usually carried out, includes the determination of acidity, saponification value, Hehner value, Reichert and Reichert-Meissl values, acetyl value, and iodine value.

Various tests are also made to determine the liability to oxidation and gumming, and colour reactions are employed to detect the presence of rosin oil, rosin acids, cholesterol, etc. Also, the presence of a soap other than an ammonia soap may be estimated by the percentage of ash contained.

Acidity. The acidity of a lubricant may be due to the presence of: (1) Free fatty acid, which exists naturally in small quantities in most vegetable oils, or may be produced by decomposition in vegetable or animal oils; (2) rosin acids present in rosin oil; (3) organic acids present in crude mineral oils; (4) free mineral acid used in refining the oil and not properly washed out during the process.

Experience has shown that the acidity due to free fatty acids in oils used as lubricants must not be greater than 4 per cent, calculated as oleic acid, otherwise there is excessive corrosion of metallic surfaces with which the oils come into contact.

The total acidity is estimated as a rule by shaking up a known weight of the oil with neutralized rectified spirit and titrating with a standard solution of caustic soda, using phenolphthalein as indicator. The acidity is expressed in terms of oleic acid, 1 cu. cm. of normal caustic soda being equivalent to 0·282 grm. of oleic acid.

Free mineral acid, which is rarely present, is determined by shaking a known weight of the oil with hot water and titrating the cold aqueous solution in a manner similar to that described above, but using methyl orange as the indicator instead of phenolphthalein. The result is expressed in terms of sulphuric acid, 1 cu. cm. of normal caustic soda solution being equal to 0·049 grm. of sulphuric acid.

The nature of the mineral acid present may be ascertained by evaporating the solution to small bulk and applying the usual tests for acids.

Saponification Value. The saponification value serves to distinguish between mineral oils on the one hand and vegetable or animal oils or waxes on the other; and as the latter in the pure state have characteristic values which have been determined by experiment, it also serves as a valuable means of identification of the individual pure oils, fats, or waxes. The saponification value may also be used to determine approximately the proportion of a mineral oil in a compound oil. It depends on the fact that when vegetable or animal oils, fats or waxes, are heated with solutions of caustic alkalies, they are decomposed or saponified with the formation of soaps and glycerol or wax alcohols.

The saponification or "alkali" value may be expressed as the quantity of caustic potash required to saponify unit quantity of oil, and is usually given in milligrams of potash per gram of fat.

Hehner Value. The percentage of insoluble fatty acids contained in a vegetable or animal oil or fat is known as the "Hehner value." Some oils contain naturally a considerable proportion of soluble and volatile fatty acids, and in the case of "blown" oils there takes place a partial conversion of the insoluble acids into soluble and volatile acids. The determination of the Hehner value, therefore, serves to detect the presence of such oils.

Reichert & Reichert Meissl Values. These values represent the proportion of soluble and volatile fatty acids, and may be required only occasionally for the detection of blown oils.

Acetyl Value. The acetyl value is useful in the analysis of castor oil and in the examination of unsaponifiable matter for wax alcohols.

Iodine Value. This test takes advantage of the fact that certain oils containing unsaturated fatty acids absorb or combine with definite proportions of iodine. The determination of the iodine value affords a very reliable means of identifying fatty oils and of ascertaining their proportions in a mixture of different oils.

Gumming. *Fatty Oils.* A good comparative test of the gumming properties of fatty oils may be made by weighing, in similar watch glasses, equal quantities of the oils to be examined and exposing them in an oven, to a constant temperature (preferably 100° C.) for a few hours. After being heated, inferior oils will be found to have increased in viscosity, and this may be observed by noting the ease with which they flow compared with a standard sample, similarly treated.

A similar test may be made by heating the oil for some time on a small sheet of plate glass, the glass having been previously heated above 100° C. to eliminate moisture. Samples must be exposed to the same constant temperature for equal times and allowed to cool.

Mineral Oils. Thickening of mineral oils may occur due to the evaporation of volatile hydrocarbons. The plate-glass test may be made, but the period of heating must be considerably greater than for the fatty oils, in order to give reliable results. Concentration of tar, rosin, or asphaltic matter may render the oil sticky when tested in this way.

CHAPTER VI

SOLID LUBRICANTS

Action of Solid Lubricants. The introduction of solid lubricant in the form of finely divided powder between dry surfaces has the effect of filling up the irregularities of the surfaces and acting as a smoothing agent. The solid friction between the surfaces is thus replaced by the lesser friction between surfaces, one (or both) of which is composed of the solid lubricant. The particles of the lubricant possess little cohesion, and seizing, therefore, is far less likely to be brought about than in the case of metallic surfaces.

Bearings operating under low pressures and speeds, and those which have to be placed in inaccessible positions where they are not likely to receive much attention, can with advantage be lubricated with solid substances.

The solid lubricant should possess the property of adhering to metallic surfaces, and should be capable of producing a smooth surface.

Common Solid Lubricants. The most common solid lubricants are graphite, mica, talc, and flowers of sulphur. Of these, graphite, mica, and talc possess a tough, flaky structure which enables them to resist pressure without disintegration. Graphite exists also in an amorphous form, and in this form it can easily be crushed into a fine powder.

Graphite. Graphite is found in the natural state in both the flaky and amorphous forms, and can be so refined as to contain 99·5 per cent pure carbon. In the variety known as No. 1340, the particles are less than $\frac{1}{300}$ in. in diameter. Dr. Acheson manufactures a colloidal graphite, in which the graphite is disintegrated into extremely fine particles and diffused in either water or oil. The products are known as the " Aquadag " and " Oildag " brands respectively, and it is estimated that the size of the particles of graphite in these products is only one-thousandth that of the particles in No. 1340 graphite.

Mica. All micas are complex silicates containing potassium, aluminium, and magnesium. They are found in the natural state, and are characterized by the ease with which they split into thin laminae.

The mica used for lubricating purposes is usually that which flakes off when large blocks are being split up for the many purposes for which this substance is used.

Talc. Talc occurs naturally in flaky, compact masses. It has a greasy touch and is very soft; it varies in colour from pearly- or silvery-white in the soft varieties to green in the harder forms. Soapstone and French chalk are varieties.

Flowers of Sulphur. This is a fine crystalline powder, occasionally used for treating hot bearings.

Addition of Solid Lubricants to Lubricating Oils. When graphite is mixed with an oil for lubricating purposes, the immediate effect is to increase the frictional resistance between the opposed surfaces.

This effect, however, is only temporary unless an excess of solid lubricant is present, in which case the accumulation of the solid particles tends, in some measure, to act as an abrasive. After continual running, the coefficient of friction is reduced, depending upon the ease with which the solid attaches itself to the surfaces to form a smooth coating. It is not, however, in the running or kinetic friction where one usually expects an improvement, but in the starting or static resistance. As stated above, the solid lubricant improves the condition of the surfaces by filling up the irregularities and pores, and it is the smoothness of the surface which is the chief factor governing the ease of starting under load.

If surfaces remain at rest under load for sufficient time to allow the oil film between them to be squeezed out, the surfaces come into metallic contact. Any subsequent motion must, in its initial stages, take place without lubrication, incurring, therefore, the risk of abrasion and seizure. The presence of a solid lubricant prevents the metallic contact and at the same time reduces the effort required at starting.

Graphite and Oil Mixtures. Experiments have been made on the Daimler-Lanchester worm gear-testing machine (p. 35) with mixtures of graphite and oil. The results of the tests show that the mineral oils are improved by the graphite when used either in the flaky form or as "Oildag." The frictional loss in the gear is reduced to a value almost equal to that obtained for the animal and vegetable oils; and, further, the critical temperature* ceases to be well marked, although a falling off in efficiency of the gear occurs above a certain temperature. One point brought out by the tests, however, is that slightly more abrasion of the bronze wheel was noticed when the flake graphite was used.

For many purposes, the use of a mixture of flaked graphite with oil would result in much saving of lubricant and, indeed, in more efficient lubrication. In steam cylinder lubrication, a considerable saving of oil may be effected, resulting in a reduction in the quantity of oil reaching the boilers. "Aquadag" has been used successfully for this purpose, and this opens up the possibility of using the exhaust steam, which in this case is free from oil, for heating, drying, or washing purposes. In the mechanically-operated lubricator designed by Mr. L. Chapman, and patented by Messrs. Chapman & Knowles, a mixture of fine graphite and glycerine is used for cylinder lubrication. The graphite is impregnated with hydrocarbons insoluble in glycerine before mixing, and the result is a mixture in which the particles of graphite remain in suspension, and which can be pumped into the cylinder in the usual manner.

The use of graphite and graphite oil mixtures for internal combustion engines is under investigation, and no definite conclusions have yet been reached as to their value.

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* Mineral oils when tested in the Daimler-Lanchester worm gear testing machine show a characteristic falling off in efficiency above a definite temperature depending on the nature of the oil. This temperature has been termed the "critical" temperature by the author for the reasons explained on pp. 77–79.

Graphite mixtures may with advantage be used in gear boxes, for driving chains, and many other similar purposes where there is sufficient splashing of the oil to keep the graphite in suspension.

An increasing amount of evidence goes to show that by the use of solid lubricants with lubricating oils, a considerable saving can be effected, and this is no doubt due to the fact that the frictional resistance of the rubbing surfaces is decreased by their increased smoothness.

CHAPTER VII

RECENT DEVELOPMENTS IN LUBRICATION

RECENT developments in lubrication include improvements in bearings, and extended knowledge concerning the lubricating properties of particular lubricants and the advantages derived from adding fatty to mineral oils. The progress achieved in each of these directions is reviewed in this chapter.

Ordinary Thrust Block. In the ordinary type of multi-collar thrust bearing, the rubbing surfaces are forced together by the load and always remain parallel to each other.

A number of collars is used in order to reduce the intensity of the pressure on the oil film to a value which experience has taught must not be exceeded.

In practice, however, it is somewhat difficult to ensure that each individual collar of the bearing carries a correct proportion of the load and, as slight errors in adjustment cause the load intensity to be greatly increased locally, the bearing as a whole is a constant source of trouble and, in fact, has become notoriously so.

Unless the lubricant is supplied at a pressure great enough to keep the rubbing surfaces apart, the lubricant will be squeezed out, allowing metallic contact to take place.

At whatever point of the bearing the oil is applied, it spreads out from that point over a large surface and is finally exuded through openings, the area of which exceeds many times the area of the oil inlet. As the bearing surfaces remain always parallel to each other, the oil film must of necessity be of the same thickness throughout ; the oil, therefore, cannot be supplied at a rate sufficient to maintain a continuous film between the surfaces, a defect which it is not possible to remedy by any special design of oil grooves.

Michell Thrust Block. In Chapter III, the lubrication of an inclined plane was discussed, and it is on the principle there explained that the Michell thrust block is designed.

In this thrust bearing the stationary surface is divided into a number of segments, each of which is supported in such a manner as to enable it to rock over until the angle between it and the moving surface is that necessary for the maintenance of a pressure oil film.

In order that the bearing may adapt itself to variations of speed, load, and lubricant, the segments are allowed a liberal degree of freedom to rock.

Figs. 19 and 20 show two methods of supporting the bearing sectors ; it will be noticed that the pivot point or line, as the case may be, is somewhat behind the centre of area of the segment.

As in this bearing the solid friction is entirely replaced by fluid friction, it is only to be expected that the coefficient of friction of the bearing as a whole is but a small fraction of that of ordinary thrust bearings.

The frictional resistances depend on the speed, the area of the bearing surfaces, and on the viscosity of the oil used. The lubricant should, therefore, have as low a viscosity as possible, consistent with maintaining a perfect oil film.

As the work lost in the bearing is due solely to shearing of the lubricant, the temperature of the latter increases. In some instances, the radiation of heat from the bearing-casing is sufficient to keep the oil at a reasonable temperature; but bearings are fitted with oil-cooling systems in which either the oil is circulated, or the bearing is run in an oil bath provided with water-cooling pipes.

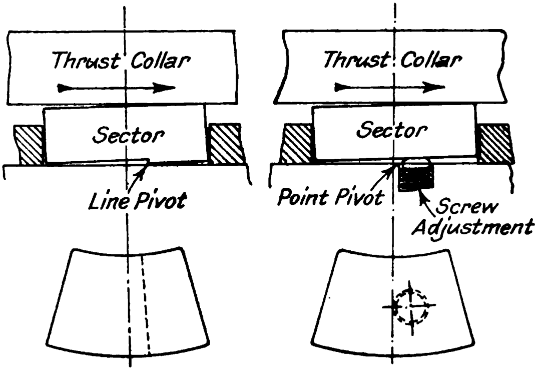

FIG. 19.—LINE PIVOT. FIG. 20.—POINT PIVOT.

Supports for Sectors in Michell Thrust Bearing.

As the result of a large number of tests, it has been demonstrated that the Michell thrust bearing will safely carry a load of 500 lb. per sq. in. of bearing surface, the coefficient of friction being 0·002. Usually, however, the bearing is designed to work at pressures between 200 and 300 lb. per sq. in. The ordinary thrust bearing cannot usually be loaded over 50 lb. per sq. in., and its frictional resistance is at least ten times that of the Michell block.

The theory underlying the design of this bearing is fully explained by A. G. M. Michell in an article on " The Lubrication of Plane Surfaces " published in *Zeitschrift für Mathematik und Physik*, vol. 50, pp. 97-155.

Table II gives the results of a series of tests carried out on a Michell thrust bearing by Messrs. Michell Bearings, Ltd., Newcastle-on-Tyne, and Fig. 21 shows a dismantled thrust block made by this firm.

TABLE II

TESTS ON A MICHELL THRUST BEARING

Pressure on Bearing Surfaces lb./sq. in.	R.P.M.	Speed ft./sec.	Temp. of Oil ° F.	Coefficient of Friction.	
				Observed.	Calcu.ated.
187	1,400	13·44	48	0·0142	0·017
187	1,400	13·44	105	0·00475	0·0063
373	1,400	13·44	115	0·00304	0·0037
560	1,400	13·44	123	0·00236	0·00269
746	1,400	13·44	130	0·00177	0·0021
1,000	1,400	13·44	137	0·00142	0·00172

Messrs. Broom & Wade (High Wycombe), manufacturers of the Michell bearing, under license from the patentees, Michell Bearings, Ltd., give the following details of a test made for the British Admiralty—

Thrust . . .	20,000 lb.
Intensity of load .	400 lb. per sq. in.
Speed . . .	460 r.p.m.
Coefficient of friction	0·0015

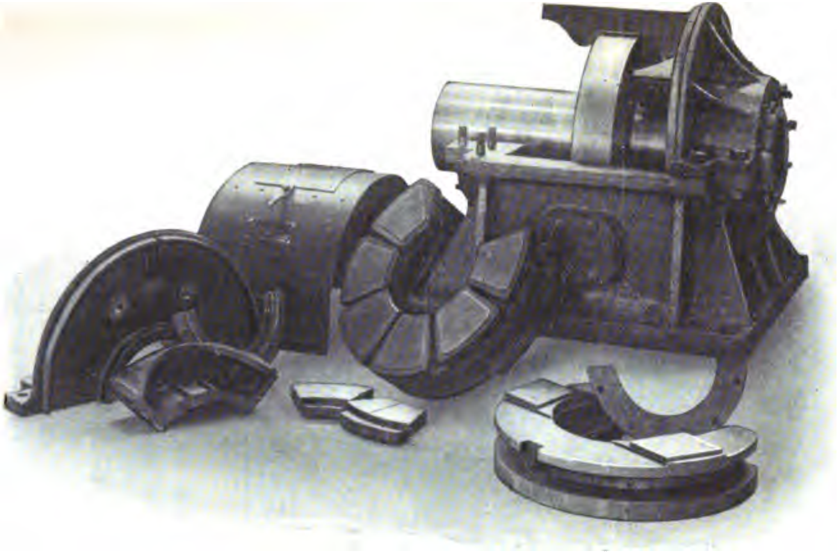

FIG. 21.—MICHELL THRUST BLOCK FOR RECIPROCATING ENGINE OR GEARED TURBINE.

The Michell Journal Bearing. The essential feature of the Michell journal bearing * is the subdivision of the bearing brasses into a number of sectors, each pivoted at its back so that it may be free to tilt at a small angle to the journal. The principle involved is the same as that utilized in the Michell thrust bearing, viz., the formation of a pressure film of oil between the rubbing surfaces.

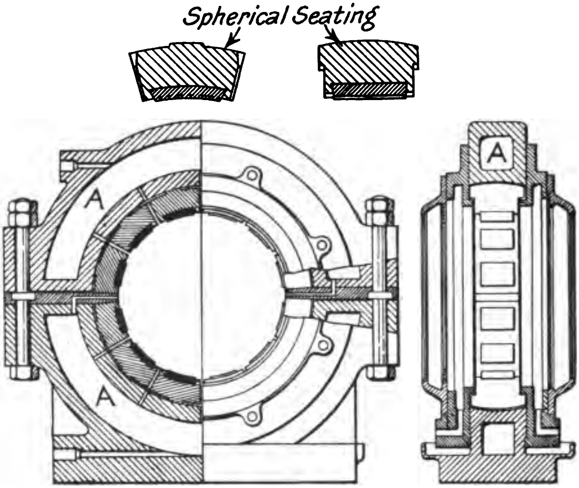

FIG. 22.—MICHELL JOURNAL BEARING.

Fig. 22 shows an experimental bearing in which there are twelve sectors. On reference to the transverse cross-sectional view, it will be seen that the bearing blocks bear against a spherical seat, thus allowing the bearing as a whole to swivel about its centre. Suitable stops are provided to overcome the tendency of the sectors to rotate with the shaft. The housing is formed hollow (as shown at A), and the oil is fed under pressure through holes in the spherical seating opposite to the space between the bearing blocks.

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* Described in Paper read by T. H. Newbigin, A.M.I.C.E. Section G, British Association, 1916.

Tests on a bearing of 12-in. diameter gave the results shown in Table III. Each run was continued until the temperature of the bearing assumed a constant value.

TABLE III

TESTS ON 12-IN. MICHELL JOURNAL BEARING

Bearing Pressure: Lb./sq. in.	R.P.M.	Surface Speed. Ft./Min.	Temperature of Oil		Coefficient of Friction.
			Inlet. ° F.	Outlet. ° F.	
220	615	1,930	84	101	0·0054
600	618	1,940	71	93	0·0034
900	1,320	4,140	76	117	0·0034

Fig. 23 shows an adaptation of the Michell journal bearing for a vertical shaft. The bearing surfaces consist of four "brasses," shown in detail at A, which are provided on the back with spherical surfaces at a and b struck from the same centre. These spherical surfaces are cleared, as shown in the plan view at f, in order that the brasses may tilt through a small angle and allow oil to enter between them and the shaft.

A cone, C, serves to hold the brasses in position in the housing and also affords a means of adjusting the running clearance.

Oil which is contained in the reservoir, *G*, revolving with the shaft, enters a nozzle, *D*, and is forced up an oil channel to the top of the bearing. A small valve at the top of the channel controls the supply. After passing through the

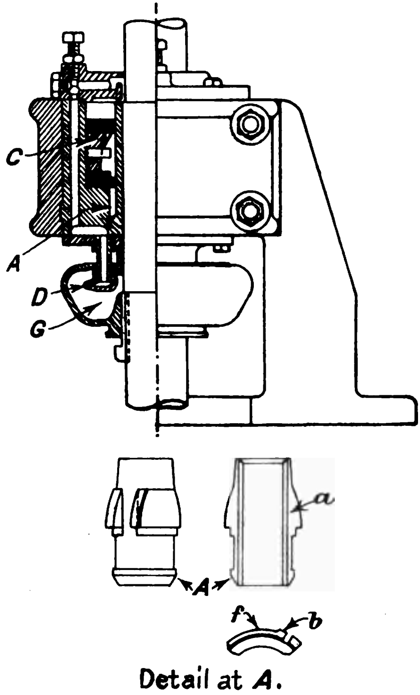

FIG. 23.—MICHELL BEARING FOR VERTICAL SHAFT.

bearing, the oil again enters the reservoir, and thus a continuous circulation is secured so long as the shaft is running.

Experiments on the Lanchester Worm-gear Testing Machine. The author has carried out experiments on this machine at the National Physical Laboratory with the object of establishing the difference in lubricating properties of lubricants derived from various sources. The tests were made for the Committee of Lubricants and Lubrication for the Department of Scientific and Industrial Research.

The lubricants selected for the work included oils of mineral, vegetable, and animal origin. These oils were not specially purified for the tests, but were commercial samples of average quality.

At the commencement of each test, the worm-gear box, oil tank, and connecting pipes were thoroughly washed out with petrol or other oil solvent, a final washing being made with a small quantity of the lubricant to be tested.

Weights corresponding to the required torque were hung on to the load pan and the machine, driven by a petrol motor, was run up to the desired speed. The machine was allowed to run for a few minutes before the belt was adjusted and the load applied, the oil meanwhile circulating through the gear-box. The temperature having become steady, the belt was tightened and the load applied. The gear-box was then balanced by adjusting the position of the slider on which the weights were hung, and the position as indicated by the micrometer attachment was noted, together with the readings of the thermocouples. The belt-tightening arrangements had continually to be adjusted slightly, in order to keep the load lifted and the gear-box gently oscillating about its level position. The load being thus applied constantly, the loss of power in the box showed itself by a slow rise in temperature of both the gear-box and the oil under test. The rate of rise in temperature was governed somewhat by the rate of flow of oil which was allowed. When the load was small and the normal running temperature of the gear low, heat was applied to the oil tank, in order to raise the apparatus to any higher temperature which was desired.

When the load was high, sufficient heat was generated in the gear-box to raise the temperature up to 100° C.

Careful tests were made to determine the effect of the efficiency of the gear due to circulating the oil in the box in the manner described. It was thought that churning the oil would affect the efficiency, but the results of comparative tests with maximum and minimum oil supply did not appreciably differ, owing no doubt to the very smooth internal construction of the gear casing and the absence of pockets and webs which could obstruct the free movement of the lubricant.

In the first series of tests made, the load was such that the mean value of the distributed pressure on the worm-wheel teeth was 1½ tons per sq. in. This value is calculated on the assumption that the worm is in contact with the wheel over the whole face of a tooth at once. Although a convenient method of defining the tooth pressure, this assumption very greatly underestimates the actual intensity of the load. The area of contact between worm and wheel is small, contact taking place practically along a line, and the load intensity is, therefore, very considerable. It has been shown that the bronze worm-wheel teeth actually flow under the enormous pressures to which they are subjected, the wheel increasing slightly in diameter over the tips of the teeth.

It must be observed that in a worm gear any part of the worm is only in contact with a particular tooth of the wheel for a short period, and both the worm and wheel are continually presenting to each other freshly lubricated surfaces. It is possible, therefore, that there is some wedging action of the lubricant between the opposed surfaces, a thin film being formed which is not entirely squeezed out during the short period for which contact endures. It would appear, therefore, that although the film is necessarily a thin one because of the great intensity of pressure, viscosity will be a factor in the lubrication, although not perhaps an important one, the lubricating quality of a thin film depending on its oiliness.

It was found that of the oils tested, the results of the tests on mineral oils showed certain characteristics not shown by the oils of animal and vegetable origin.

Variation of Efficiency with Temperature. Usually the efficiency of a gear for a mineral oil was lower than when a fixed oil was used. The efficiency values plotted on a temperature base (Fig. 24) show in the case of mineral oils a more or less sudden falling off at a particular temperature,

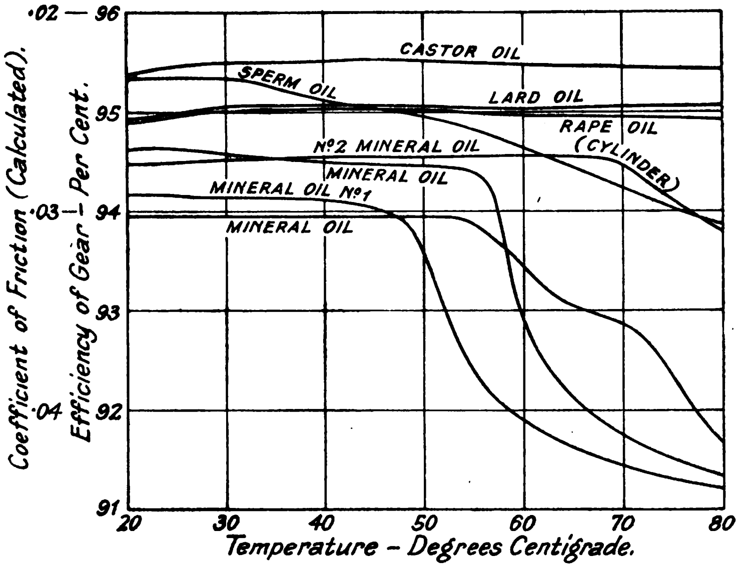

FIG. 24.—EFFICIENCY OF WORM GEAR, WITH VARIOUS OILS AT VARIOUS TEMPERATURES.

(Tests on Daimler-Lanchester machine, N.P.L.)

called the "critical temperature." In the case of sperm oil, a falling off was observed, but it occurred gradually. When the critical temperature of the mineral oils was reached, the machine became difficult to operate, the efficiency falling too rapidly with a small rise in temperature to allow steady results to be obtained. With some mineral oils, violent oscillations were set up in parts of the machine, indicating the critical point even when the fall in efficiency was neither sudden nor of great amount.

In Fig. 24, the variation in efficiency of the gear with temperature is plotted for various oils. Castor oil gives practically a constant efficiency of 95·5 per cent, corresponding to a coefficient of friction * of 0·0225; and lard and rape oil give an efficiency of 95 per cent, the coefficient of friction being 0·025.

In the case of the mineral oil No. 1, the efficiency maintains a nearly constant value of 94·1 per cent (coefficient of friction 0·0295) until a temperature of 46° C. is reached. Above this temperature, the efficiency rapidly falls off, but again assumes an

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* The value of the coefficient of friction, μ, between the surfaces of worm and wheel teeth is calculated from the equation

$$\varepsilon = \frac{\tan \theta}{\tan (\theta + \phi)}$$

where $\mu = \tan \phi$
ϕ = angle of friction,
θ = angle of worm,
ε = efficiency, as a fraction (*not* per cent).

This equation, when the efficiency is high, reduces to—

$$\mu = \frac{1 - \varepsilon}{2}.$$

approximately steady value of 91·5 per cent ($\mu = 0{\cdot}0425$) above 70° C., at which temperature the test was stopped, as there was a possibility of damaging the worm gear. The same characteristic curve of efficiency-temperature is shown by all the mineral oils tested and, although the critical temperature is not well marked in some of the curves, it was always apparent during the tests by the behaviour of the machine.

Viscosity and Efficiency. Table IV gives the results of a few tests on this machine, and it will be seen that the viscosity of the oil is little or no guide to its value as a lubricant under the conditions of these tests. When the increase in viscosity due to pressure is taken into account, and the results are plotted on a viscosity base instead of a temperature base, there is still no definite relation between the internal friction of the lubricant and the efficiency of the worm gear.

Experiments similar to those described above were made at a tooth load of 2 tons per sq. in. and at a speed of 1,500 r.p.m., and similar differences between the oils were observed.

Effect of Mixing Rape Oil with Mineral Oil. For this investigation, Mineral Oil No. 1 (Fig. 24) was selected as being typical. A test was first made on the mineral oil, which was then allowed to cool to atmospheric temperature. The test was then recommenced and, when the machine was running steadily and a perfect balance of the gear-box obtained, a quantity of rape oil corresponding to 5 per cent mixture was added to the oil tank and stirred vigorously with the mineral oil. In a short time, the balance was disturbed and a readjustment had to be made, showing an

TABLE IV

VISCOSITY OF OIL AND EFFICIENCY OF WORM GEAR AT VARIOUS TEMPERATURES

Tests made on Daimler-Lanchester Worm-gear Testing Machine. (N.P.L.)

Temperature : ° C.	CASTOR OIL.		LARD OIL.		MINERAL OIL, NO. 1.		MINERAL OIL NO. 2.	
	Viscosity.*	Efficiency of Gear, %.	Viscosity.*	Efficiency of Gear, %.	Viscosity.*	Efficiency of Gear, %.	Viscosity.*	Efficiency of Gear, %.
20	7·5	95·4	0·89	94·9	1·64	94·2	19·6	94·5
30	4·1	95·5	0·55	95·1	0·84	94·2	7·6	94·5
40	2·2	95·5	0·37	95·1	0·54	94·1	3·8	94·5
50	1·1	95·5	0·26	95·1	0·33	93·6	2·1	94·5
60	0·68	95·5	0·18	95·0	0·22	91·9	1·25	94·5
70	0·45	95·5	0·14	95·1	0·16	91·5	0·81	94·4
80	0·29	95·4	0·11	95·1	0·09	91·2	0·53	93·8

* Viscosity in absolute units (C.G.S.)

alteration in the efficiency due to the addition of the rape oil.

The results of these tests are shown in Fig. 25. The very slight rise in efficiency by adding the rape oil will be noticed, but what is more important than this insignificant rise is the raising of the critical point of the lubricant from 46° C. to 55° C.

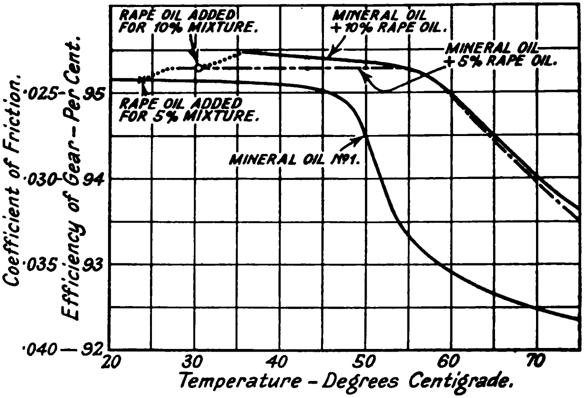

FIG. 25.—EFFICIENCY OF WORM GEAR LUBRICATED BY MINERAL OIL, WITH AND WITHOUT RAPE OIL.

Tests on Daimler-Lanchester machine (N.P.L.).

The curve for a 10 per cent mixture shows a further small increase in efficiency due to the additional rape oil, and practically the same critical temperature as for the 5 per cent mixture. This effect of raising the critical temperature was found to be the same for other oils of mineral origin, the critical point in one case rising 18° C.

These tests demonstrate what for many years has been found in practice to be the case, that the addition of a small quantity of fatty oil considerably improves a mineral oil. Tests on mixtures up to 25 per cent rape oil were made, but it was found that the addition of 2½ per cent of the fatty oil was almost as effective as the 25 per cent mixture.

Further tests were made on the addition of graphite, and the general conclusions reached are noted on pp. 64–66.

Improving the Lubricating Properties of Mineral Oils by Addition of Fatty Oils. The tests described above show that the addition of a small percentage of fatty (animal or vegetable) oil to a mineral oil has the effect of reducing in some degree the frictional losses.

In 1916,* Irving Longmuir carried out important experiments on surface films, and came to the conclusion that the adsorption of liquids at the surface of a solid body was the result of unsaturated chemical forces at the surface. The adsorbed layer he estimated to be only one or two molecules thick. He explained the fact that animal and vegetable oils spread over the surface of water, while mineral oils do not, as being due to the presence of an " active group " in the molecules in the former case and its absence in the latter.

Dr. H. S. Allen, M.A., has pointed out † that the most efficient oil will be that which will most readily spread over a metallic surface to form a lubricating layer, and that its " oiliness " will depend not on its viscosity or compressibility, but on "the chemical forces called into play between the active part of the oil molecule and the solid surfaces of the bearing." This being the case, the value of the lubricant will depend on its nature and on the material of which the surface is formed.

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* *Journal Amer. Chem. Soc.*, Vol. xxxviii, p. 2221, 1916.

† *Journal Soc. Chem. Ind.*, 1920

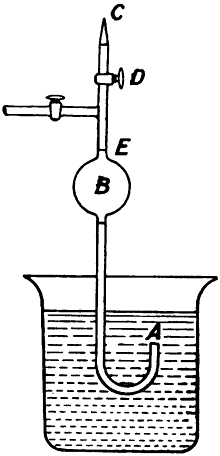

FIG. 26.—SOUTHCOMBE AND WELLS'S APPARATUS FOR DETERMINING SURFACE TENSION OF OILS AGAINST WATER.

Messrs. Southcombe & Wells have made numerous experiments in order to ascertain the reason for the superior lubricating qualities of the fatty oils over the pure mineral oils.*

They have measured the surface tension of oils against water, using the apparatus shown in Fig. 26. This consists of a U-shaped capillary tube provided with a ground glass orifice at *A* and a bulb at *B*. The long limb of the U-tube is provided, above the bulb, *B*, with a stop-cock, *D*, the tube at its upper end being drawn down to a fine orifice at *C*. The apparatus is supported with the ground glass orifice, *A*, under the surface of water contained in a beaker.

The apparatus is filled up to the mark *E* with the oil to be tested and, when the stop-cock is opened, a very small flow of oil is obtained from

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* Discussion on "Lubrication," Physical Society of London, November, 1919.

A. The number of drops produced by a given volume of the oil is counted, the number being inversely proportional to the interfacial tension, against the water, of the oil under test.

A large number of tests showed that the measured tension was very much less in the case of the fatty oils than in the case of the mineral oils.

Taking the value of the tension of the mineral oils against water as 100, the value for rape oil was found to be 68, for lard oil 73, and for cocoanut oil 59.

Similar experiments on oils with and without fatty acids gave the results shown in Table V.

TABLE V

COMPARATIVE SURFACE TENSION OF OILS AGAINST WATER

Oil.	Free Fatty Acid Content. %	Comparative Tension Value.
Pure mineral	nil	100
98% mineral, 2% fatty acid .	1·9	80
97% mineral, 3% fatty acid .	2·8	78
Olive oil	2·2	80
Olive oil	4·5	72
Olive (neutral)	0·1	92
Rape oil	2·5	76
Rape (neutral)	0·15	93

The effect on the lubricating value of a mineral oil treated with fatty acids is demonstrated by some experiments carried out by Mr. L. Archbutt on a Thurston oil-testing machine. For the purpose of these tests, the machine was run at a load of 270 lb. per sq. in. and at the low speed of

7 ft. per min., as, under these conditions, the effect of "oiliness" is best demonstrated.

The mineral oils were treated with "Tonicol," a preparation containing 25 per cent of free fatty acid, and supplied by Messrs. The Henry Wells Oil Co., Ltd. The tests showed that the reduction in the friction effected by the addition of 2½ per cent of "Tonicol" was equal to that obtained by adding 10 per cent of ordinary commercial rape oil.

Mr. Archbutt made similar tests on mixtures of mineral oil and rape oil fatty acids, the results of which are given in Table VI. Other tests on rape and mineral oils gave the results shown in Table VII.

TABLE VI

COMPARATIVE FRICTION OF MINERAL OIL WITH VARIOUS PERCENTAGES OF RAPE OIL FATTY ACIDS

Mineral Oil.	Rape Oil Fatty Acids. %	Comparative Value of Friction. %
100·0	nil	100
99·5	0·5	76
99·0	1·0	68
98·0	2·0	64

As a result of their experimental work, Messrs. Southcombe and Wells came to the conclusion that a satisfactory explanation of "oiliness" was to be found in the *lowering of the interfacial tension between oil and water* by the presence of free fatty acid. This conclusion, however, is hardly justified in view of Archbutt's data in

Table VII, which show that neutral rape oil is almost as effective in reducing the friction as is acid rape oil.

TABLE VII

COMPARATIVE FRICTION OF MINERAL OIL, AND NEUTRAL AND ACID RAPE

Oil.	Comparative Value of Friction.
Mineral . . .	100
Neutral rape . .	64
Acid rape, 2·4% free fatty acid . .	58

There is no doubt, however, about the friction-reducing value of free fatty acid, as will be seen from the data in Table VIII, obtained from experiments carried out with the Deeley oil-testing machine on mild steel surfaces.

TABLE VIII

STATIC COEFFICIENT OF FRICTION BETWEEN MILD STEEL SURFACES WITH VARIOUS OILS

Description of Oil.	Static Coefficient of Friction.
Pale mineral oil (paraffin base) . .	0·227
Pale mineral oil + 2% " Tonicol " . .	0·160
Pale mineral oil + 4% " Tonicol " . .	0·153
Red mineral oil (paraffin base) . .	0·191
Red mineral oil + 2% " Tonicol " . .	0·136
Red mineral oil + 4% " Tonicol " . .	0·120

The treatment of mineral oil with " Tonicol," in the manner described, has been termed by Messrs. Southcombe & Wells, the " Germ Process."

CHAPTER VIII

EXAMPLES OF LUBRICATION AND LUBRICATORS

Lubrication of Steam Cylinders. The magnitude of the internal frictional losses of a steam engine depends largely upon the degree of lubrication of its piston and valves. Indeed, the steam consumption of an engine running without internal lubrication may be as much as 10 to 20 per cent greater than when ample lubrication of these parts is provided.

With internal lubrication of a condensing engine, however, very great care must be taken to ensure that the lubricating oil is not returned to the boiler. The presence of the oil in the boiler may lead to considerable local overheating, resulting, perhaps, in the collapse of furnace tubes or other serious damage ; and for this reason many marine engines are run without internal lubrication, in spite of the loss of power which results not only from the increased friction, but from the leakage of steam past the pistons.

Exhaust Steam Oil-separators. The dangers due to oil in the boiler feed water may be partially overcome by the employment of an oil-separator. This apparatus consists usually of a large vessel containing a series of baffle plates and vanes against which the exhaust steam impinges. The comparatively large volume of the vessel ensures a reduction of the steam velocity, resulting in partial condensation of the steam and the precipitation of most of the oil present. The baffles are so arranged that the steam is sent round a sinuous path and comes into contact with a series of vanes on which the oil is deposited, and from which it drains away into gutters and finally to the base of the vessel, from which it is drawn off.

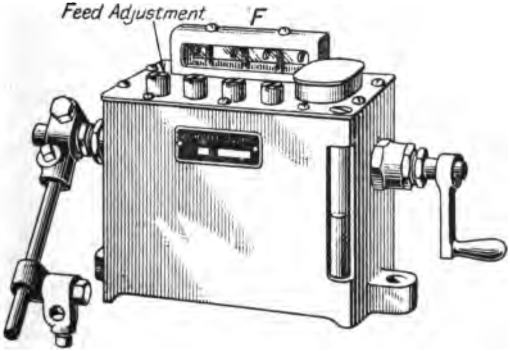

FIG. 27.—"MANZEL" FORCE-FEED OIL PUMP. EXTERNAL VIEW.

In cases where oil-separators are not used, the feed-water is purified either by filtering it through asbestos fibre and similar substances, or by chemical means, the latter method being used where feed-water softening plants are installed.

"Manzel" Force-feed Oil Pump. A modern type of cylinder lubricator is that shown in Figs. 27 and 28, and known as the "Manzel" lubricator. It consists essentially of a small force-pump, with a regulating plunger which delivers oil to a second pump, by which it is, in turn, forced through a check valve into the oil pipe line. Referring to the sectional view in Fig. 28, the regulating plunger, *R*, is reciprocated by the small eccentric on the shaft, *O*, through the crosshead, *C*. This plunger sucks the oil from the reservoir (in which the whole pump is immersed) through a strainer, and delivers it through the piston valve shown. By adjusting the position of the plunger with respect to the crosshead, the quantity of oil pumped can be

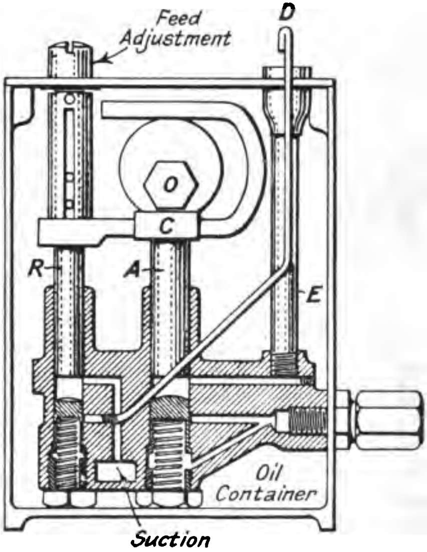

FIG. 28.—"MANZEL" FORCE-FEED OIL PUMP. SECTIONAL VIEW.

regulated within fine limits. The oil is delivered through the tube, *D*, to the vertical pipe, *E*, and the passage of the oil, in drops, can be observed through the glass cover, *F* (Fig. 27). The main plunger, *A*, also connected to the crosshead, *C*, thus delivers only that quantity of oil which it receives from the regulating pump and which can be observed.

The small piston valves employed are so fitted as to be very easily accessible. The check valve is designed to prevent the steam from backing into the oil lines, and to keep the pipe filled with oil.

The pump is operated by means of a lever actuating a ratchet and pawl mechanism, on the eccentric shaft, the lever being connected to some oscillating part of the engine.

The operating mechanism is enclosed in the oil reservoir, and is thus protected from dust and dirt.

The lubricator is fitted with a small handle for use before starting the engine or when a larger supply than the normal feed is required whilst the engine is running. The quantity of oil delivered may be adjusted from one drop to every four strokes of the plunger to about 12 drops per stroke.

Application of Lubricant in Steam Cylinders. The common method of introducing the oil into a steam cylinder is either directly through the cylinder walls or through the steam chest. In either case, it is questionable whether the oil gets distributed over the surfaces of the cylinder and the valves and, in all probability, a considerable part of it is swept out by the exhaust steam before it performs its proper function.

In order to overcome this difficulty, the oil is sometimes admitted to the steam pipe, near the steam chest, through a tube of special design termed an "atomizer." This tube is so fixed that it protrudes to the centre of the steam pipe, where the oil comes into contact with the steam moving at the greatest velocity. Under these conditions, the oil is broken up into minute particles and dispersed through the steam, and is thus carried to the walls of the cylinder and to the valve seatings. This method has been applied successfully by Mr. T. C. Thomsen to steam engines of many types.

"Scientific" Mechanical Lubricator. Another form of sight-feed lubricator, introduced by Mr. A. Kirkham, is illustrated in Fig. 29. As in the case of the "Manzel" apparatus, there is a flooded pump operated from a shaft which is oscillated by a motion-rod from the engine. The example illustrated is a four-feed lubricator of 1 gall. capacity.

Attached to the rocking shaft are two levers or pendulums with special projections. The ends of the pump plungers are pressed hard against these projections by means of springs which operate the plungers on the suction stroke. By this arrangement the pump mechanism is kept free from backlash, as is essential if uniformity of oil supply is to be maintained.

The arrangement of the pump valves can be seen on reference to the figure. On the suction stroke, the oil passes through the cock, *G*, up through the inlet valve, *V*, down the sloping oil passage and into the pump barrel. During the

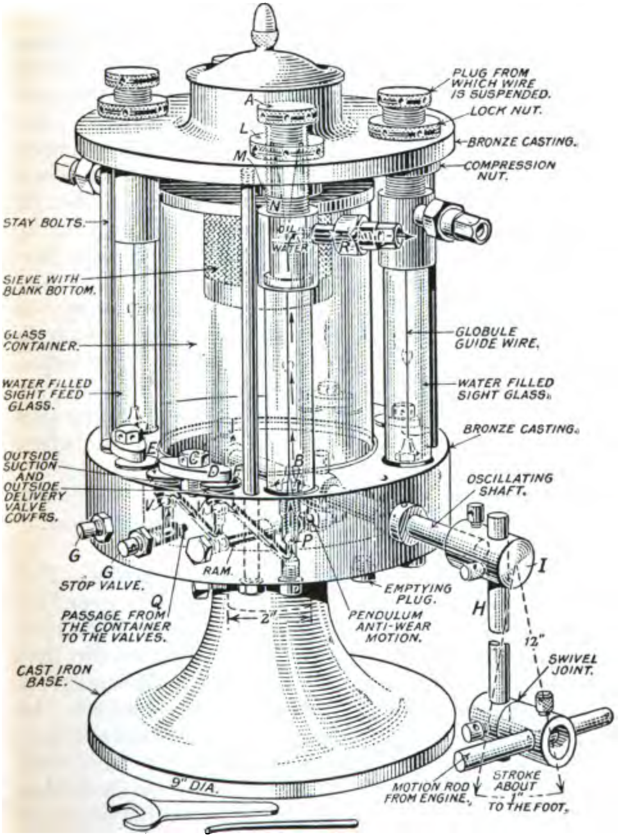

Fig. 29.—“Scientific” Mechanical Lubricator.

pressure stroke, the oil is forced up through the second ball valve, V_1, down the second sloping oil passage, and up through the valve, *P*, which is held on its seat by a spring. The oil then rises through the nozzle, *B*, and up through the water column, whence it passes through the check valve, *R*, to the engine. A wire stretched vertically from the nozzle, *B*, confines the oil to the centre of the water column, thus preventing the sight glasses from becoming oily and permitting the passage of the lubricant to be seen clearly. The stroke of the plungers can be adjusted in order to regulate the supply of oil as required, and this operation may be performed while the lubricator is working.

The lubricant is supplied to the apparatus through the top cover, and is strained before entering the glass container.

As will be seen from the figure, the valves are easily accessible for the purposes of inspection.

Lubrication of Automobile Gear Boxes. The lubrication of the ordinary automobile gear-box may be effected simply and efficiently by allowing the teeth of the gears on one shaft to dip into troughs in which the level of the lubricant is maintained. Failing this, it will generally be sufficient to put into the box such a quantity of oil that the larger gears dip into it; this, in a gear box of the ordinary type, will be such that the level of the oil is about half-way between the bottom of the box and the shaft axis. It is a great mistake either to put too much oil into the gear box, or to use a very thick oil or a grease.

In 1914 the author carried out some experiments * on a 32-h.p. Leyland gear-box with the object of determining the effect on the efficiency of power transmission due to the viscosity and also the quantity of lubricant used. A sectional view of the gear-box is shown in Fig. 30.

In these experiments, the gear box was supported in a tilting frame in such a manner that it could oscillate through a small angle about the axis of the driving shaft. The torque on the driving shaft and the reaction torque on the gear box were measured, the power transmitted being absorbed by a friction brake. This method of test is much superior to the usual method of determining the ratio of "input" to "output," as it determines the loss of power directly and not as a difference of two large quantities which may be themselves so much in error as to establish considerable doubt as to their difference. The

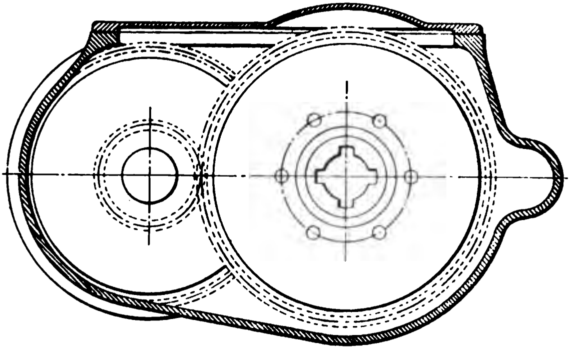

FIG. 30.—CROSS-SECTION OF AUTOMOBILE GEAR BOX.

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* "National Physical Laboratory Report, 1914–15."

efficiency determinations were estimated to be accurate to within 0·2 per cent.

The results of the experiments showed that considerable loss of power resulted when the level of the oil in the gear-box was higher than about midway between the bottom of the box and the shaft centre. (*See* Fig. 30.) The maximum efficiency, 97·5 per cent, was obtained with the oil just below this level. With the box about half-full of oil, level with the shaft centre, the efficiency was 94 per cent. When the oil level was midway between the shaft centre and the top of the largest gear, the efficiency obtained was only 90 per cent. Further addition of oil rapidly lowered the efficiency, until, when the box was quite full, the efficiency fell to 75 per cent. When the box was more than half full, the temperature of the oil rose rapidly, but the figures for efficiency quoted above were all taken when the temperature was 110° F.

This simple experiment shows that there is a loss of power of 25 per cent when the box is full of oil, compared with only 2½ per cent, or one-tenth of the amount, when the box is one-quarter full. Further experiments at different speeds and loads, and with different oils, showed conclusively that the power loss above 2½ per cent was due entirely to the work done on the oil by the gearing; in fact, the gear-box acted as an admirable fluid-friction brake when filled with oil.

So far as the gearing alone is concerned, a thin oil of good lubricating value is the most desirable lubricant to use. Generally, however, an oil of medium viscosity is chosen because of its being less liable to leak past the shafts and also because of its tendency somewhat to deaden the "ring" of the gears.

The use of very thick oil or grease is particularly undesirable, especially if the gears are running at high speeds, because of the viscous friction set up. In the case of very thick greases, the gears at high speed simply carve a path in the grease and get rid of all they initially retain by squeezing during meshing, and by centrifugal force, leaving the gear teeth bereft of lubricant.

The loss of power in spur-gear boxes, when too much oil is used, is in a great measure due to the vigorous stirring of the oil produced by the two sets of gear wheels rotating in opposite directions, and by resistance to free motion of the lubricant set up by internal webs and pockets. In the case of a worm-gear box, the power lost when the box is full of oil is not appreciably greater than when a small quantity of oil is used, provided the inside of the box follows the contour of the worm and wheel, and is free from webs and projections.

Live Axle Lubrication. For the reason already explained in the case of gear boxes, the rear axle casing of a motor car should not be crammed with lubricant, particularly if the drive be made through bevel gearing. Sufficient oil of medium viscosity to ensure a wetting of the rubbing or rolling surfaces is infinitely preferable to a back axle casing filled with oil or grease. The use of forced lubrication, by fitting a plunger pump and supplying oil under pressure to all bearing parts and to the gear teeth in a spray, has been advocated, and there is no doubt that this is an excellent way of lubricating the live axle.

Universal Joints, Steering Gear, Suspension, etc. In the design of universal joints and similar parts, it is too often assumed, when the lubrication is considered, that it is only necessary to provide access for oil or grease to some part of the spindle or bearing requiring lubrication. Consequently, it is not unusual to find universal joints running in oil, but with no provision made for the access of the oil to the bearing pins, which are often heavily loaded. Care should be taken that the part to be lubricated is not robbed of oil by the action of centrifugal force, as would be the case in a universal joint which is not lubricated from the shaft centre.

The lubrication of steering-gear knuckle-joints and such parts as spring shackles is often effected by fitting grease caps, but these are liable to get knocked off or to be ignored, and are consequently falling into disfavour.

Usually, oil is preferable to grease as a lubricant, and it has been suggested, as a solution of the problem of lubricating the coach spring, to totally enclose the whole spring and its shackles in a casing with flexible ends and fill the casing with oil.

The lubrication of steering knuckles has been effected by fitting ball and socket joints rendered oil-tight by spring caps, and flooded, through suitably placed oil-ways, by oil contained in the connecting tubes.

Ball and Roller Bearings. As the area of contact between a ball or a roller and its race is very small, the pressure intensity is correspondingly large, and it is doubtful whether any oil film can be sustained intact under such conditions.

A correctly designed ball or roller bearing performs its work with the very minimum of deformation of either of the rolling surfaces; and, unless the designed load is exceeded in practice, there will be almost pure rolling and no rubbing or slipping. However, deformation of the ball or roller and the race must exist in a small degree, and lubrication is necessary in order to obtain the most satisfactory results from this class of bearing.

Generally, a pure mineral grease is found to be the best type of lubricant for ball and roller bearings. The greatest care should be taken that the grease is really pure and free from animal and vegetable fats, which, although they are very good lubricants, are liable to develop free acid and corrode the bearing surfaces. Greases containing graphite, talc, and similar substances are also unsuitable for these bearings, especially if, as is commonly the case, they contain small particles of silica, which rapidly causes wear.

The author has seen ball bearings which for six months had been kept in store and "preserved" by smearing over them a thick yellow grease of unknown composition, yet of a type in common use. When cleaned, both the balls and races were deeply pitted and had all the appearance of having been thoroughly rusted-up.

In gear boxes and similar mechanisms, where oil is plentiful, the bearings become flooded; and, provided the oil is well strained and free from grit, etc., the lubrication of ball bearings is quite satisfactory.

The housings of ball and roller bearings should be designed to exclude effectually water and dirt, otherwise a lapping action will be set up and the bearings will rapidly be rendered useless.

Tests at the National Physical Laboratory on Hoffmann ball bearings compared with ordinary shaft bearings gave the results shown in Table IX. The journal diameter was 2½ in., and the total load 2,300 lb. The length of the plain bearing was 5 in., and that of the Hoffmann ball bearing, 1¾ in. overall.

TABLE IX

COEFFICIENT OF FRICTION IN PLAIN AND BALL BEARINGS

Revolutions per Minute.	Plain bearing with Oil-ring Lubricator.	Hoffmann Ball Bearing.
80	0·015	0·0013
130	0·014	0·0014
250	0·013	0·0015
500	0·012	0·0016
Starting torque	770 lb.-in.	0·9 lb.-in.

In these tests, the running friction of the ball bearings was less than one-tenth part of that of the plain bearing, and the static friction 1/850th part.

Lubrication of Petrol Motors. The lubrication of a petrol motor may be effected by forcing the oil under pressure to the parts requiring lubrication, by so arranging splash-troughs that the oil is splashed from them to these parts, or by a combination of the splash and pressure systems.

Splash System. In this system, the oil is transferred from a reservoir to the crank chamber of the engine by means of a small pump, the latter being operated by hand or mechanically. The level of the oil in the crank chamber, known as the "splash level," is such that the connecting rods, usually provided with small projections for the purpose, dip into the oil and splash it over the moving parts of the engine. Suitably placed troughs or gutters are provided which lead the oil thus distributed to lubricating holes in the bearing caps, connecting rod ends, and similar places.

In this system, the oil is not circulated, the loss being made up from time to time by the pump as required.

Pressure System. The pressure or force-feed system consists in circulating the oil, by means of piping and oil passages, to the crankshaft bearings, crank pins, gudgeon pins, camshaft and cams, and to the timing and other gears.

The oil is forced by means of a pump to the main bearing caps, whence it passes through the crankshaft and crank webs, which are suitably bored and drilled for the purpose, to the connecting rod big-end bearings, and thence by pipe or through the hollow connecting rods to the gudgeon pins.

In some cases, the gudgeon pins are formed hollow; and the oil, passing through their ends, serves to lubricate the cylinders and pistons, but the latter are usually lubricated by the oil thrown about by the connecting rods.

The stream of oil exuding from the various bearings finds its way finally into a sump at the bottom of the crank case, from which it again passes, through a strainer, to the circulating pump.

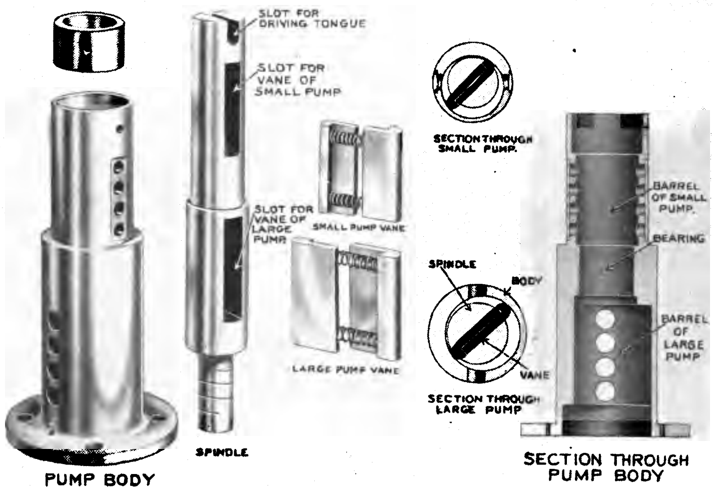
SLOT FOR DRIVING TONGUE
SLOT FOR VANE OF SMALL PUMP
SLOT FOR VANE OF LARGE PUMP
SMALL PUMP VANE
LARGE PUMP VANE
SECTION THROUGH SMALL PUMP.
SPINDLE
BODY
VANE
SECTION THROUGH LARGE PUMP
BARREL OF SMALL PUMP.
BEARING
BARREL OF LARGE PUMP
PUMP BODY
SPINDLE
SECTION THROUGH PUMP BODY

FIG. 31.—DETAILS OF OIL PUMP FOR " WOLSELEY " HISPANO-SUIZA AERO ENGINE.

Combined Systems. Combinations of these two systems are in common use. Thus, in some cases, the crankshaft bearings and crank pins only are supplied with oil under pressure, the lubrication of the gudgeon pins and other parts depending upon the oil thrown off the connecting rod ends. In other cases, the crankshaft bearings alone are pressure-fed, and the connecting rod big-ends dip into splash troughs and scatter oil to lubricate the rest of the engine.

Lubrication of " Wolseley " Hispano-Suiza Aero Engine. The lubrication system of this engine is taken as a typical example of aero engine practice. The oil recommended for use is pure castor oil. It is supplied under pressure by a rotary pump, and is delivered directly to the crankshaft and big-end bearings, to the camshaft bearings and cams, and by means of a jet to the reduction gears with which this engine is fitted. The cylinders and pistons are lubricated by oil thrown off by the connecting rod ends.

The oil is supplied under a pressure of 120 lb. per sq. in., the pressure being controlled and regulated by an adjustable relief valve. The consumption of oil varies from 10 to 12 pints per hour for a 200-h.p. eight-cylinder engine of 120 mm. bore and 130 mm. stroke.

The engine is fitted with two vane-pumps running in tandem and shown in Fig. 31. The lower pump drains all the oil from the crank-case and delivers it to a supply tank, from which it is again drawn by the upper pump, forced through a strainer, and thence into the main supply pipe. The oil passes under pressure to

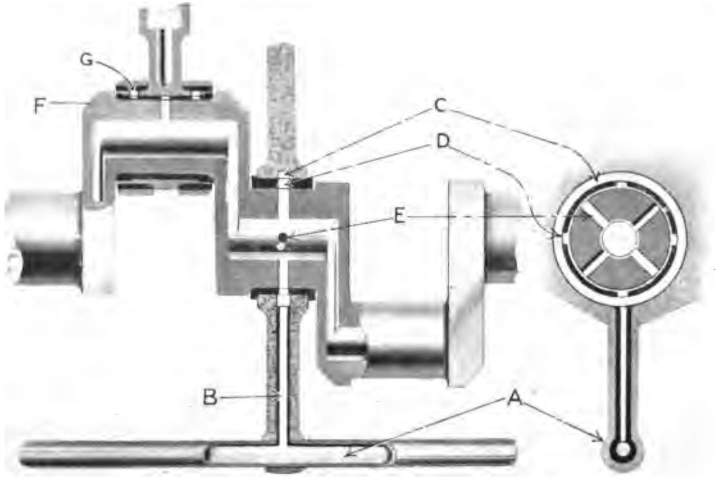

FIG. 32.—AERO ENGINE CRANKSHAFT WITH BIG-END BEARINGS.

the underside of each crankshaft bearing, as shown in Fig. 32. An annular groove cut round the inside of the bearing housing enables the oil to pass round the outside of the bearing brasses, and through the holes which are drilled in them, to the crankshaft. The brasses are shown on Fig. 33.

Connecting Rod Big-ends. Fig. 32 shows the oil passages through the webs by which the pressure oil passes, after flooding the crankshaft bearings, to the big-ends. In this engine, two big-ends work on the same crank pin, one being forked to enshroud the other and taking its bearing on it. This is shown at *G* (Fig. 32), the dark portions representing the white metal bearing surfaces, the inner surface being drilled in order to allow oil to pass to the bearing of the forked rod.

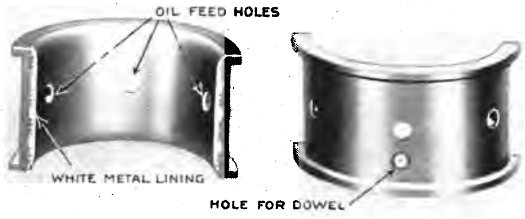

FIG. 33.—AERO ENGINE CRANKSHAFT BEARINGS.

Gudgeon Pins. The small-end of the connecting rod is drilled at the top to allow oil splashed up inside the piston to find its way into the small-end bearings. Oil is also collected from the cylinder walls by a groove round the piston, which connects with the centre of the gudgeon pins, the latter being drilled with small holes to enable the oil to reach the small-end bush.

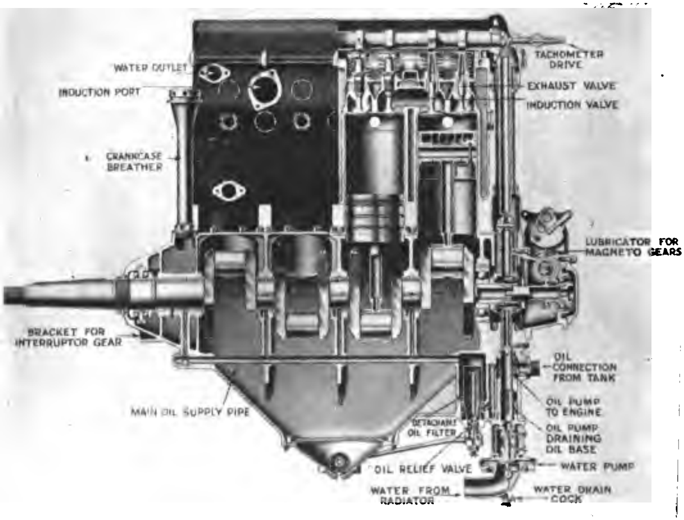

FIG. 34.—"WOLSELEY" HISPANO-SUIZA AERO ENGINE.
Part longitudinal section showing lubrication system.

Camshaft and Cams and Gears. Oil under pressure is supplied through the hollow camshaft to the camshaft bearings and through a small hole in the "rest" portion of each cam to the valve mechanism generally.

The oil, in draining back to the gear case, lubricates the valve-timing and magneto gears, which in this case are bevel and skew gears respectively.

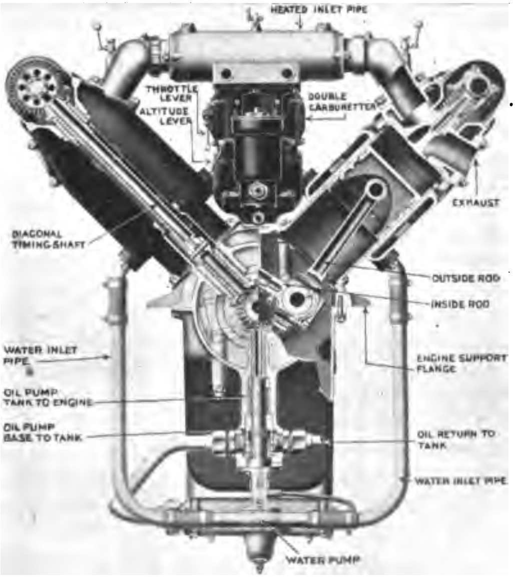

FIG. 35.—"WOLSELEY" HISPANO-SUIZA AERO ENGINE.
Transverse section showing lubrication system.

The driving gears are lubricated by a jet of oil impinging on the teeth as they enter into engagement.

The complete lubricating system is shown in Figs. 34 and 35.

Lubrication of Turbine Gearing. The very high peripheral speed of turbine gear wheels renders the ordinary bath system of gear-box lubrication unsuitable, owing to the heavy viscous drag of the lubricant on the wheel surfaces. The energy which would be expended by the gears in moving through the oil at high speed, and in scattering it inside the case, would be such as to necessitate the rapid circulation of a very large supply of lubricant, in order to keep the temperature down to a reasonable limit.

Experiments described in Chapter VII have shown the desirability of keeping the working temperature of a mineral oil as low as possible, in order that the critical temperature may not be exceeded.

In the lubrication of turbine gears, the oil is sprayed over the surfaces of the teeth just as they are entering into gear, and the oil, falling afterwards into a tank beneath, is drawn off by a pump, cooled and used again.

Generally, a moderately thin mineral oil is used, but great care should be taken that the lubricant does not easily emulsify with water, because the presence of the latter in the gear case cannot well be avoided. The use of a heavy viscous lubricant is neither desirable nor necessary.

Figs. 36 and 37 show the lubricating system of a turbine gear manufactured by Messrs. David Brown & Sons, Ltd., of Huddersfield. Oil is supplied under pressure to all the shaft bearings and the spraying nozzles by means of the pump, *G*. It is drawn into the pump from the main tank beneath the gear case and forced through the oil cooler and strainer, after which it is distributed by various pipes to all parts of the gear requiring lubrication. The oil is delivered at a pressure of 10 lb. per sq. in., the pressure being indicated on a gauge and regulated by means of a relief valve.

The design of the nozzles employed is such that a maximum spread of the lubricant is obtained for the particular oil and feed pressure chosen. It has been found much more satisfactory to spray the lubricant over the whole width of the gear teeth than to rely on lubrication from one or two narrow jets.

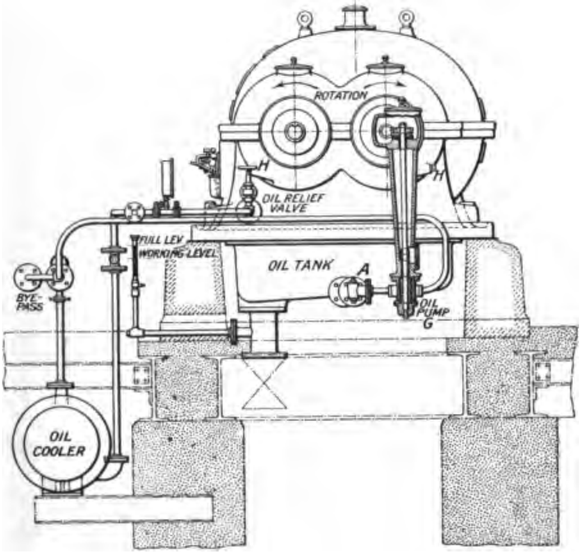

FIG. 36.—GENERAL ARRANGEMENT OF LUBRICATION SYSTEM FOR TURBINE GEARS.

The oil from the spraying nozzles, after being used, falls to the bottom of the case. Similarly, the oil to the bearings enters the bearing on one side and passes out at the other, after which it returns to the case and thence to the tank.

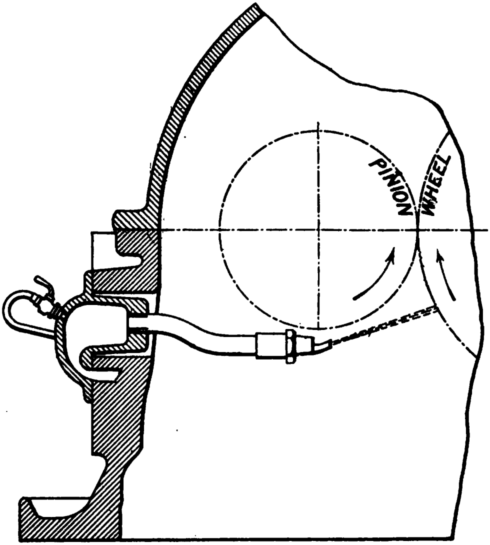

FIG. 37.—CROSS-SECTION SHOWING SPRAY PIPE FOR LUBRICATION OF TURBINE GEARS.

In order to obtain the temperature of the return oil from the bearings, bosses are cast on the bearing housing. Each boss receives a tube which dips into the oil and contains a small quantity of mercury in which the thermometer bulb is immersed. The manufacturers recommend that a temperature of 140° F. should not be exceeded.

The lubricating system should be examined periodically, the cooler and strainer being cleaned out and the water which settles from the oil being driven off.

It is essential that the oil used should be kept below its critical temperature, otherwise there may be serious loss of power, and possibly injury to the gear, due to the reduced lubricating quality of the oil at high temperatures. Tests on turbine gear oils have shown that some of the lubricants used are inferior to others at low temperatures, but much superior at the working temperature of the gear.

BIBLIOGRAPHY

FOR further information on lubricants and the theory and practice of lubrication, the reader may be referred to the undermentioned publications—

Friction and Lost Work in Machinery and Millwork, by R. H. Thurston. (Chapman & Hall.)

Practice of Lubrication, by T. C. Thomsen. (McGraw-Hill.)

Lubrication and Lubricants, by Archbutt and Deeley. (Griffin.)

Petroleum and Its Products, by Sir Boverton Redwood. (Griffin.)

Lubricating and Allied Oils, by E. A. Evans. (Chapman & Hall.)

Oils, Fats, and Waxes, by Fryer and Weston. (University Press.)

Oils, Fats, and Waxes, by Lewkowitsch. (Macmillan.)

Practical Treatise on Animal and Vegetable Oils and Fats, by W. T. Brannt. (Sampson, Low & Co.)

Chemistry of Petroleum, by Tinkler and Challenger. (Crosby Lockwood.)

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